COMPRESSED AIR

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A TREATISE ON THE PRODUCTION TRANSMISSION AND USE OF COMPRESSED AIR

BY

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PREFACE TO SECOND EDITION

In preparing a second edition of this book, its character as an elementary treatise on the principles of Compressed Air has been maintained throughout. To bring the subject up-to-date, however, the greater portion of the chapter on Transmission has been re-written. Modern formulas were introduced with examples of their application. The important effect of altitude differences on Compressed Air installations has been illustrated by reference to new equations in Article 4 and by the example under Article 97. A new Table VIII, containing the principal Compressed Air formulas was substituted for the old table that has become superfluous. The obsolete Hurricane valve in Article 147 has been replaced by a description of the modern Ingersoll-Rogler valve.

Altogether, a general revision of the original text and the addition of new material brings the second edition of the book abreast of modern theory and practice.

THEODORE SIMONS.

BUTTE, MONTANA,
December, 1920.

PREFACE TO FIRST EDITION

This treatise is intended to give the student and the general reader such an insight into the natural laws and physical principles underlying the production, transmission and use of compressed air, as shall enable him to comprehend the operation of the various appliances employed for this purpose and to judge of their merit.

No attempt has been made to present in this book an extensive description of all the existing types of compressors or of the countless appliances using compressed air. The author's chief aim was to provide the student, who is interested in technical questions concerning the operation as well as the construction of compressors and air engines, with a background of understanding that will enable him, not only to solve the many theoretical problems connected therewith, but to make independent research into the seemingly unlimited possibilities of compressed air. The territory still unexplored is vast and full of promises to the intrepid explorer who enters the field with a thorough knowledge of all the truths discovered, as well as the pitfalls encountered, by those who have gone before him.

The numerous, carefully selected problems constitute what the author believes to be one of the strong features of this book. If ever any doubt lingers in the student's mind as to the meaning of certain principles or laws presented in the text and their practical application, a numerical problem will, as a rule, remove the doubt and make clear the meaning. Moreover, such problems make the student familiar with actual quantities, never revealed by mere formulas; quantities which are often startling to the uninitiated and impress him with the practical value of such formulas more forcibly then the mere text can do.

The author has endeavored to bring the work well within the comprehension of the average technical student who has a sound knowledge of the elements of algebra, physics and mechanics. Higher mathematics were used sparingly and only when they led to a simpler solution of certain problems. To the advanced reader some of the deductions contained in the book

may appear unnecessarily lengthy. It has been the writer's experience, however, that many of the difficulties encountered by students arise from a misunderstanding of facts which, although perfectly obvious to one who has mastered the subject, remain nevertheless obscure to the beginner unless explained from various points of view and by analogy with facts already familiar to him.

In preparing this treatise the writer has made free use of the rather scattered and by no means voluminous literature on the subject of compressed air. His debt to all who have labored in this field before him can hardly be acknowledged adequately by the mere mentioning of their names. He has therefore refrained from referring in the text to such names but wishes to express in this preface his gratitude to all authors and investigators from whose writings he has drawn both inspiration and information.

Throughout the preparation of this work the author had the untiring assistance of President C. H. Bowman of the Montana State School of Mines, whose constructive criticism and suggestions, based on a vast theoretical knowledge and practical experience, were of inestimable value. To him the author is indebted to an extent that the mere mentioning of the fact can scarcely requite.

For permission to use diagrams, illustrations, tables and other data, contained in the bulletins of manufacturers, the author is indebted to the following firms: Allis Chalmers Company, Ingersoll Rand Company, Nordberg Manufacturing Company, Norwalk Iron Works Company, Sullivan Machinery Company, The Laidlaw-Dunn-Gordon Company and Union Steam Company.

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PART I

THE PRODUCTION OF COMPRESSED AIR

COMPRESSED AIR

CHAPTER I

THE PHYSICAL PROPERTIES OF AIR. DEFINITION OF TERMS USED IN THE DISCUSSION OF COMPRESSED AIR PROBLEMS

1. Composition of Air.—Air is chiefly composed of the elements oxygen and nitrogen. By weight the proportions are about 23 parts of oxygen and 77 parts of nitrogen. By volume the proportions are about 21 parts of oxygen and 79 parts of nitrogen.

2. Weight of Air.—By actual measurement it has been found that 1 cu. ft. of air at atmospheric pressure at sea level and at 60° Fahr. weighs 0.0764 lb. Since the density of air changes with the pressure and with the temperature, it follows that the weight of a given volume of air varies with pressure and temperature. How this weight can be computed when pressure and temperature are known is shown in Articles 21 to 24.

3. Atmospheric Pressure.—Since air has weight, it is evident that the enormous quantities of air that constitute the atmosphere must exert a considerable pressure upon the earth.

By experiment the atmospheric pressure at sea level, with the barometer at 30 in. and a temperature of 32° Fahr., has been found to average 14.7 lb. per square inch above vacuum.

4. Air Pressures at Varying Altitudes and Temperatures.—The solving of many compressed-air problems requires a knowledge of the pressure at any point of a vertical column of free or compressed air. Without having recourse to actual measurements, these pressures may be calculated as follows:

Let Fig. 1*A* represent an air column having a base 1 sq. in. in area and a height of h feet.

Let P_1 and P_2 = absolute pressures per square inch at bottom and at top of column, respectively.

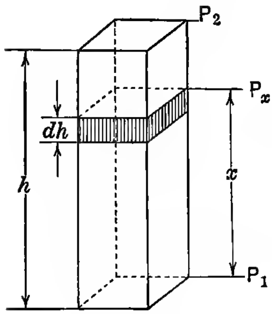


FIG. 1a.

P_x = absolute pressure at a point x ft. above bottom, in pounds per square inch.

w_x = weight in pounds of one cubic foot of air at point x.

P_a = atmospheric pressure at sea level (14.7 lb.).

w_a = weight in pounds of one cubic foot of atmospheric air at sea level and at an absolute temperature T_a.

Then the absolute pressure at a point dh ft. below x will be

$$P_x + dP_x \text{ lb. per sq. in.}$$

in which dP_x = weight in pounds of shaded air column.

But $$dP_x = \frac{1}{144} dh w_x \text{ lb.}$$

whence $$dh = 144 \frac{dP_x}{w_x} \tag{1}$$

From Art. 22 $$\frac{w_x}{w_a} = \frac{P_x}{P_a}$$

whence $w_x = w_a \frac{P_x}{P_a}$. Introducing in (1) gives

$$dh = 144 \frac{P_a}{w_a} \frac{dP_x}{P_x}$$

whence $$h = 144 \frac{P_a}{w_a} \int_{P_2}^{P_1} \frac{dP_x}{P_x} = 144 \frac{P_a}{w_a} \log_e \frac{P_1}{P_2} \tag{2}$$

From Art 22 the weight of one cubic foot of atmospheric air at sea level and at an absolute temperature T_a is

$$w_a = \frac{39.804}{T_a} \text{ lb.} \tag{3}$$

Introducing (3) in (2) and using common logarithms, gives

$$h = 144 \frac{14.7 T_a}{39.804} \times 2.302 \ (\log P_1 - \log P_2)$$

whence $$\log P_2 = \log P_1 - \frac{h}{122.4 \text{ (degree F.} + 461)} \tag{4}$$

For an average temperature of 60° Fahr. equation (4) becomes:

$$\log P_2 = \log P_1 - 0.0000157h \qquad (5)$$

whence $\log P_1 = \log P_2 + 0.0000157h \qquad (6)$

and $$h = \frac{\log P_1 - \log P_2}{0.0000157} = 6370 \log \frac{P_1}{P_2} \qquad (7)$$

Example 1.—Atmospheric pressure P_1 at sea level is 14.7 lb. What is atmospheric pressure P_2 at an elevation of 5000 ft. above sea level and at a temperature of 60° Fahr.?

$$\log P_2 = \log 14.7 - \frac{5000}{122.4(60+461)}$$

$$P_2 = 12.27 \text{ lb. per. sq. in.}$$

Example 2.—By barometric observation the atmospheric pressure at a certain mine was found to be 11.24 lb. per sq. in. Average temperature 60° Fahr. What is the elevation of the mine above sea level?

From equation (7)

$$h = 6370 \log \frac{14.7}{11.24} = 7400 \text{ ft.}$$

Example 3.—A compressor at a line, located 5500 ft. above sea level, furnishes compressed air for air-drills at the bottom of a shaft 2800 ft. deep. A pressure gage in the pipe line, near the shaft bottom reads 60 lb., when the air is not moving.

(*a*) What is the absolute pressure corresponding to this gage pressure?

(*b*) What should be the reading of a pressure-gage in the pipe line at the collar of the shaft and at the same instant?

Solution (*a*). Elevation of shaft bottom above sea level is $5500 - 2850 = 2650$ ft.

Atmospheric pressure at shaft bottom

$$\log P_{2650} = \log 14.7 - 0.0000157 \times 2650$$

whence $$P_{2650} = 13.36 \text{ lb.}$$

Added to 60 lb. gives absolute pressure of compressed air at shaft bottom

$$P_1 = 60 + 13.36 = 73.36 \text{ lb. abs.}$$

Solution (*b*). Atmospheric pressure at shaft collar:

$$\log P_{5500} = \log 14.7 - 0.0000157 \times 5500$$

whence $P_{5500} = 12.05$ lb.

Absolute pressure of compressed air at shaft collar:

$$\log P_2 = \log 73.36 - 0.0000157 \times 2850$$

whence $P_2 = 66.2$ lb. abs.

Gage pressure $66.20 - 12.05 = 54.15$ lb. gage.

In well-regulated practical operations such readings are taken frequently and at various points in the transmission line. A marked discrepancy between the actual and the computed readings indicates trouble in the pipe line such as leaks or careless waste of air. The cause of the discrepancy is then traced to its source and the proper remedies are applied before much harm is done.

5. General Effect of Heat on Air.—Heat has a tendency to increase the volume of air, that is, to expand it. If air at outside temperature is confined within a closed cylinder, and then heated, the result of this tendency to expand may be two-fold:

1. If the cylinder is tightly closed at both ends, and if the walls are strong enough to resist deformation, the volume of air will remain constant and its pressure will increase.

2. If in the upper end of the cylinder we insert a piston which is free to move in the cylinder and has a certain weight, it will descend in the air-filled cylinder until its weight is balanced by the pressure of the confined air.

If now the air in this cylinder is heated, it expands and the piston will start upward, and will stop when the expansion has ceased; in this case, the load of the piston and consequently the pressure of the air have remained the same as before heating, but the volume has increased.

The effect of heat upon the air in the cylinder is, therefore, in the first case, to increase its pressure under constant volume; and in the second case, to increase the volume under constant pressure.

Reversely, if we take a closed cylinder full of hot air and allow

it to cool, the volume of this air will, of course, remain the same, but its pressure will fall gradually, until it becomes the same as it was before heating.

In a similar way, if we allow the heated air in the cylinder with its piston to cool, the volume of air confined under the piston will shrink, and the piston will gradually descend to the point where it was before the air was heated, the pressure, of course, remaining constant.

The effect of abstracting heat from the air in the cylinder is, therefore, in the first case, to decrease the pressure under constant volume; and in the second case, to decrease the volume under constant pressure.

6. Specific Heat and The British Thermal Unit.—The specific heat of a body is the ratio between the amount of heat required to raise the temperature of that body 1 degree and that required to raise the temperature of an equal mass of water 1 degree.

In engineering problems the British Thermal Unit (B.T.U.) is usually employed as the unit of heat. It is the quantity of heat required to raise the temperature of 1 lb. of water 1° Fahr. Thus, the specific heat of water is 1, and the specific heat of any substance is the number of B.T.U.'s required to raise the temperature of 1 lb. of that substance 1° Fahr.

By a law of thermodynamics, heat and mechanical energy are mutually convertible, and heat requires for its production, and produces by its consumption, a definite amount of work for each thermal unit.

The mechanical equivalent of the British Thermal Unit has been found to be very close to 778 ft.-lb. This value will be used throughout this treatise.

Thus: $$1 \text{ B.T.U.} = 778 \text{ ft.-lb.}$$

7. Specific Heat of Air at Constant Volume.—If heat is applied to air contained in a closed vessel, the air is said to be heated under constant volume. In this case the number of heat units required to raise the temperature of 1 lb. of air by 1° Fahr. is the specific heat of air at constant volume.

Expressed in B.T.U.'s it is: $C_v = 0.1689$ B.T.U.'s.
Expressed in foot-pounds it is: $K_v = C_v \times 778 = 131.6$ ft.-lb.

8. Specific Heat of Air at Constant Pressure.—If heat is applied to air in a cylinder having a movable piston under a constant

external pressure, the volume increases, and therefore work is done in pushing the piston out against the external pressure. The number of heat units required in this case to raise the temperature of 1 lb. of air by 1° Fahr. is the specific heat of air at constant pressure.

Expressed in B.T.U.'s it is: $C_p = 0.2375$ B.T.U.'s.
Expressed in foot-pounds it is: $K_p = C_p \times 778 = 184.8$ ft.-lb.

C_p is greater than C_v owing to the extra heat required to do the work of moving the piston against the external resistance, in addition to raising the temperature of the air. Theory indicates, and experiment shows, that the excess of heat $(K_p - K_v)$ required in the latter case is equal to the amount of work done by the air in expanding against a constant pressure.

The above specific heats are for dry air. They will be used throughout this treatise, although the presence of moisture in the air slightly modifies these values.

9. Absolute Zero.—Direct experiment, in which air at constant pressure was exposed to various temperatures, has shown that the volume which it occupies at a temperature of 32° Fahr. increases or decreases by 1/493 of this volume for each increase or decrease of 1° Fahr.

From this it follows that air heated under constant pressure to a temperature of boiling water (212° Fahr.) has increased in volume by $\frac{212-32}{493} = \frac{180}{493} = 0.366$ or 36 per cent. of the volume it occupied at 32° Fahr. The same air at 493° below the freezing point of water or 461° below 0° Fahr. would have shrunk by $\frac{32+461}{493} = \frac{493}{493}$ of its volume, or by that volume itself. The temperature at which this is assumed to take place is called *"the absolute zero."* For ordinary compressed air problems it is taken as 461° below 0° Fahr. This value is used throughout this treatise.

10. Absolute Temperature.—Absolute temperature is the temperature above the absolute zero. It is usually designated by T while temperatures in degrees Fahr. are designated by t. At 60° Fahr. the absolute temperature T is $60° + 461° = 521°$. At 0° Fahr. the absolute temperature T is $0° + 461° = 461°$. At $-30°$ Fahr. the absolute temperature is $-30° + 461° = 431°$.

11. Gage and Absolute Pressures.—Ordinary gages register pressures above atmosphere. Thus, if the air gage of a compressor shows 80 lb. pressure, it indicates that the pressure of the compressed air is 80 lb. per square inch above the pressure of the atmosphere. To find the absolute pressure of air compressed at sea level to 80 lb., we must add 14.7 to the gage reading; thus $80+14.7=94.7$ lb. absolute. The pressures indicated by the gage are called gage pressures; pressures above vacuum are called absolute pressures. To obtain absolute pressure at any altitude, add atmospheric pressure at that altitude to the gage pressure.

From Table VI atmospheric pressure at 10,000 ft. elevation is 10.07 lb. per square inch. Hence a gage pressure of 100 lb. at an altitude of 10,000 ft. is equal to $100+10.07=110.07$ lb. absolute pressure.

12. Free Air.—Free air is a term constantly used in dealing with problems of air compression. It is air at normal atmospheric pressure as taken into the cylinder of a compressor.

13. Isothermal Compression or Expansion of Air.—From experiment we find that heat is generated in the act of compressing air. If during compression the air could be kept at constant temperature by the abstraction of heat as fast as it was generated, the air would then be said to be compressed isothermally.

In expanding against an external resistance, the air gives up, or, to speak more correctly, converts heat into mechanical energy. If as much heat could be supplied and as fast as it is consumed, the air would be said to expand isothermally. In isothermal compression or expansion the air remains at constant temperature throughout the operation.

14. Adiabatic Compression or Expansion.—If during compression the air neither loses nor gains heat, the heat generated by the compression remaining in the air and increasing its temperature, then the air is said to be compressed adiabatically. When the compressed air is allowed to expand against an external resistance its temperature falls, and if the air during this operation receives no heat from without, it is said to expand adiabatically.

CHAPTER II

BEHAVIOR OF AIR UNDERGOING COMPRESSION AND UNDER THE APPLICATION OF HEAT

There are two fundamental laws governing the behavior of air undergoing compression and under the application of heat. These laws express the relations existing between volume, pressure and temperature.

15. Boyle's or Mariotte's Law.—The temperature remaining constant, the volume of a given weight of air varies inversely as the absolute pressure.

$$\frac{V_1}{V}=\frac{P}{P_1} \text{ or : } P_1V_1=PV$$

whence $$V_1=V\frac{P}{P_1}$$

and $$P_1=P\frac{V}{V_1}$$

in which V = volume of a given weight of air at an absolute pressure P.

V_1 = volume of the same weight of air at the same temperature and at any absolute pressure P_1.

Example.—One-hundred cubic feet of free air, compressed isothermally at sea level to 60 lb. gage will occupy a volume:

$$V_1=\frac{100\times14.7}{60+14.7}=19.68 \text{ cu. ft.}$$

Conversely, 19.68 cu. ft. of air at 60 lb. gage, when expanded isothermally down to atmospheric pressure, will occupy a volume:

$$\mathrm{V}=\frac{\mathrm{V_1P_1}}{\mathrm{P}}=\frac{19.68(60+14.7)}{14.7}=100 \text{ cu. ft.}$$

16. Boyle's Law may also be expressed as follows: The temperature remaining constant, the product of the pressure P and the volume V is a constant.

$$PV=P_1V_1=\text{constant}$$

Example.—It has been found that at sea level 1 lb. of air at atmospheric pressure and at 32° Fahr. occupies a volume of 12.387 cu. ft.

If we let P = absolute pressure in pounds per square foot

V = volume of air in cubic feet

Then for $P = (14.7\times144) = 2116.8$ lb. per square foot

and $V = 12.387$ cu. ft.

$PV = 2116.8\times12.387 = 26{,}220$, nearly.

If the pressure is increased under constant temperature to two atmospheres or 29.4 lb. absolute per square inch, the volume of the pound of air will have been reduced to one-half of the original volume. We will then have:

$P_1 = (29.4\times144) = 4233.6$ lb. per square foot

$V_1 = 12.387\times1/2 = 6.1935$ cu. ft.

whence $P_1V_1 = 4233.6\times6.1935 = 26{,}220$

and $PV = P_1V_1 = \text{constant}$

17. Charles' or Gay Lussac's Law.—If the pressure remains constant, every increase of temperature of 1° Fahr. produces in a given quantity of air an expansion of 1/493 of the volume it occupies at a temperature of 32° Fahr.

$$V_1 = V(1+at^\circ)$$

in which V_1 = volume of a given weight of air at t° Fahr. above the freezing point.

V = volume of same weight of air at freezing point (32° Fahr.).

t = number of degrees rise in temperature above freezing point.

a = coefficient of expansion $=1/493$.

Example.—One pound of atmospheric air at 32° Fahr. at sea level occupies a volume $V = 12.387$ cu. ft. At a temperature of 62° Fahr. and at the same absolute pressure it would occupy a volume:

$$V_1 = 12.387\left(1+\frac{1}{493}\times30\right) = 13.141 \text{ cu. ft.}$$

18. Charles' Law may also be expressed as follows: Under constant pressure the volume which a given weight of air occupies at different temperatures, varies directly as the absolute temperatures.

Let V = volume of a given weight of air at an absolute pressure P and an absolute temperature T.

V_1 = volume of the same weight of air at the same absolute pressure P and at any absolute temperature T_1.

Then

$$\frac{V_1}{V}=\frac{T_1}{T}$$

whence

$$V_1=V\frac{T_1}{T}$$

and

$$T_1=T\frac{V_1}{V}$$

Example.—One pound of air at 32° Fahr. and at atmospheric pressure at sea level occupies a volume $V=12.387$ cu. ft. At a temperature of 62° Fahr. and at atmospheric pressure it would occupy a volume:

$$V_1=\frac{VT_1}{T}=\frac{12.387(461+62)}{461+32}=13.141 \text{ cu. ft.}$$

Column 2 of Table II gives the volume in cubic feet occupied by 1 lb. of air at various temperatures, at sea level.

19. Another deduction may be made from Charles' Law as follows: If a certain weight of air be heated to different temperatures in a closed cylinder so that its volume remains constant, the absolute pressures vary directly as the absolute temperatures.

$$\frac{P}{P_1}=\frac{T}{T_1}$$

whence

$$P_1=P\frac{T_1}{T}$$

and

$$T_1=T\frac{P_1}{P}$$

Example.—Let absolute pressure of a volume of free air at a temperature of 62° Fahr. be 14.7 lb. per square inch. If heated to a temperature of 200° Fahr. without changing its volume, the absolute pressure of the heated air would be:

$$P_1=P\frac{T_1}{T}=14.7\times\frac{200+461}{62+461}=18.58 \text{ lb. absolute or } 3.88 \text{ lb. gage.}$$

20. Boyle's and Charles' Laws Combined.—Given a quantity (weight) of air which has a volume V, a pressure P, and a temperature T, we can change it to a condition in which its volume is V_1, its pressure P_1, and its temperature T_1. First: Change the pressure from P to P_1 under constant temperature T, then find the new volume V_n from Boyle's Law:

$$\frac{V_n}{V}=\frac{P}{P_1} \tag{1}$$

Second: Change the temperature of this volume V_n from T to T_1 under constant pressure P_1. Then find the new volume V_1 from Charles' Law:

$$\frac{V_1}{V_n}=\frac{T_1}{T} \qquad (2)$$

Multiplying equations (1) and (2) we get:

$$\frac{V_n V_1}{V_n V}=\frac{P T_1}{P_1 T}$$

Whence $$P_1 V_1 = PV\frac{T_1}{T} \qquad (3)$$

In Article 16 it was shown that for 1 lb. of air at 32° Fahr. and at atmospheric pressure:

$$PV=26{,}220$$

Substituting this value in equation (3) we get:

$$P_1 V_1=\frac{26{,}220}{32+461}\, T_1 = 53.2\ T_1 \text{ nearly} \qquad (4)$$

The equation is usually written:

$$PV=RT \text{ or } P_1 V_1 = RT_1 \qquad (5)$$

in which

P and P_1 = absolute pressures in pounds per square foot
V = volume in cubic feet which 1 lb. of air occupies at a temperature of 32° Fahr. and at atmospheric pressure (14.7 lb.)
V_1 = volume in cubic feet which 1 lb. of air occupies at an absolute pressure P_1 and an absolute temperature T_1
R = constant $= 53.2$.

From equation (5) we deduce:

$$V_1=\frac{RT_1}{P_1}$$

If this volume of air be raised 1° in temperature at constant pressure, its volume will become:

$$V_2=\frac{R(T_1+1)}{P_1}$$

The change of volume will be:

$$V_2 - V_1 = \frac{R(T_1+1)}{P_1} - \frac{RT_1}{P_1}$$

whence $$P_1(V_2 - V_1) = R(T_1 + 1 - T_1) = R \qquad (6)$$

The first term of the equation is the work done by the pound of air in expanding against a constant pressure P_1 while the temperature of the air is rising 1 degree. In Article 8 it was stated that this work is equal to the difference in the specific heats, expressed in foot-pounds.

Hence we may write:

$$R = (K_p - K_v) = 184.8 - 131.6 = 53.2 \qquad (7)$$

which is the same as the value deduced in equation (4).

Example.—One pound of air at 32° Fahr. ($T = 461 + 32$) and at atmospheric pressure P (14.7) occupies a volume $V = 12.387$ cu. ft.

If we compress this pound of air under constant temperature T to 100 lb. gage ($P_1 = 114.7$ lb.) we find the new volume V_n from equation (1).

$$V_n = 12.387 \, \frac{14.7}{114.7} = 1.588 \text{ cu. ft.}$$

If we heat this volume V_n of air, which still weighs 1 lb., from 32° Fahr. to 150° Fahr. ($T_1 = 461 + 150$) under constant pressure, we find the new volume V_1 from equation (2):

$$V_1 = 1.588 \times \frac{461+150}{461+32} = 1.967 \text{ cu. ft.}$$

Expressing P_1 in pounds per square foot $= 144 \times 114.7 = 16{,}517$ lb.

We get $$P_1V_1 = 16{,}517 \times 1.97 = 32{,}500$$

The same result could have been obtained directly from equation (4)

$$P_1V_1 = 53.2\ T_1$$

$$P_1V_1 = 53.2\ (150 + 461) = 32{,}500$$

In employing formulas (4) and (5) of this article, it must be borne in mind that the pressures are expressed in pounds per square foot, and that the quantity of air contained in the volume is 1 lb.

WEIGHT OF AIR

21. Weight of Equal Volumes of Air at Constant Pressure and Varying Temperatures.—Let a cylinder (Fig. 1 a) with a movable piston, be filled with a given weight of air at an absolute temperature T and an absolute pressure P in pounds per square inch, occupying a volume V_a.

If heat is applied to the cylinder (*a*) the air in it will expand under constant pressure and the piston will assume the position shown in Fig. 1 *b*. The weight of this air has remained the same and so has the pressure, but the volume and the absolute temperature have increased, while the density of the mass of air has decreased.

If we now cut off from Fig. 1 *b* a volume equal to the volume V_a in Fig. 1 *a* it is evident that the weight W_1 of the volume V_a in cylinder (*b*) is less than the weight of volume V_a in cylinder (*a*).

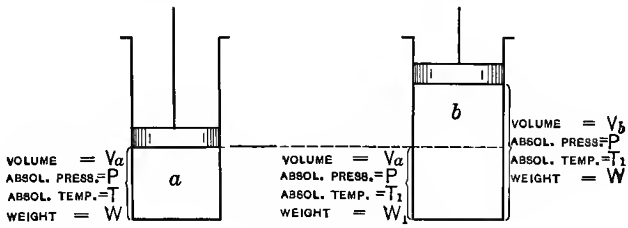


FIG. 1.

This shows that of two equal volumes of air having the same absolute pressure, the one having the higher temperature has the less weight. The exact relation may be derived from the equations in Article 18. It is stated as follows:

The weight of two equal volumes of air, having the same absolute pressure, varies inversely as the absolute temperatures.

$$\frac{W}{W_1} = \frac{T_1}{T}$$

in which W = weight of a given volume of air at an absolute temperature T

W_1 = weight of an equal volume of air at an absolute temperature T_1

22. Weight of Equal Volumes of Air at Constant Temperature and Varying Pressures.—Let a cylinder (Fig. 2 *b*) having a movable piston be filled with a given weight of air occupying a volume V_b and having an absolute pressure P and an absolute temperature T.

If we load the piston with an additional weight (*m*), the piston will descend in the cylinder to the position shown in Fig. 2 *a*. The weight of the air in cylinder (*a*) has remained

the same, and the absolute temperature is assumed to have also remained the same. But the absolute pressure has increased and with it the density of the air.

If we now cut off from the cylinder (*b*) a volume V_a equal to the volume in cylinder (*a*), it is evident that the weight of volume V_a in cylinder (*b*) is less than the weight of volume V_a in cylinder (*a*). This shows that of two equal volumes of air having the same absolute temperature, the one having the less pressure has the less weight, and *vice versa*. The exact relation may be derived from the equations in Article 15. It is stated as follows:

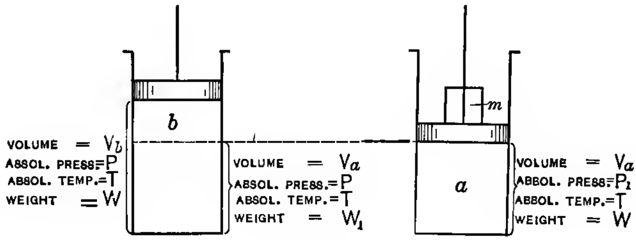


FIG. 2.

The weight of two equal volumes of air, having the same absolute temperature, varies directly as the absolute pressures.

$$\frac{W}{W_1}=\frac{P}{P_1}$$

in which W = weight of a given volume of air at an absolute pressure P

W_1 = weight of an equal volume of air at an absolute pressure P_1

23. Weight of 1 cu. ft. of Air at Atmospheric Pressure P at Sea Level and at any Absolute Temperature T_1.—According to Article 2, 1 cu. ft. of air at atmospheric pressure at sea level and at 60° Fahr. weighs 0.0764 lb.; hence the weight W_1 of 1 cu. ft. of air at atmospheric pressure but at an absolute temperature T_1 is

$$\frac{W_1}{W}=\frac{T}{T_1} \quad \text{whence} \quad W_1=W\frac{T}{T_1}=\frac{0.0764\ (60+461)}{T_1}=\frac{39.804}{T_1}$$

Thus, we find the weight of 1 cu. ft. of air at atmospheric pressure at sea level and at any absolute temperature T_1 by dividing the constant 39.804 by the absolute temperature T_1.

24. Weight of 1 cu. ft. of Air at any Absolute Temperature T_1 and any Absolute Pressure P_1.—We have as before: $W_1=\frac{39.804}{T_1}$ = weight of 1 cu. ft. of air at atmospheric pressure at sea level and at any absolute temperature T_1.

If we now designate by W_2 the weight of 1 cu. ft. of air at the same absolute temperature T_1 but at any absolute pressure P_1 we have from Art 22:

$$\frac{W_1}{W_2}=\frac{\text{atmospheric pressure at sea level}}{P_1}$$

$$\frac{W_1}{W_2}=\frac{14.7}{P_1}$$

Substituting the value of W_1 from Art. 23 we get $\frac{39.804}{T_1W_2}=\frac{14.7}{P_1}$

whence
$$W_2=2.7077\ \frac{P_1}{T_1} \qquad (1)$$

Thus, we find the weight of 1 cu. ft. of air at any absolute pressure and temperature by multiplying the absolute pressure in pounds per square inch by the constant 2.7077 and dividing the product by the absolute temperature.

Example.—The weight of 1 cu. ft. of air at 60 lb. gage and at 100° Fahr. at sea level is:

$$W_2=2.7077\ \frac{60+14.7}{100+461}=0.3602 \text{ lb.}$$

Table I gives the weight of 1 cu. ft. of air at various gage pressures and temperatures at sea level.

24*a*. When using formula (1), Article 24 for computing the weight of 1 cu. ft. of air at an elevation above sea level, it must be borne in mind that P_1 in that case is the gage pressure plus the atmospheric pressure at that elevation.

Example.—What is the weight of 1 cu. ft. of air at 60 lb. gage and at 100° Fahr., at an elevation of 8000 ft. above sea level?

From Table VI, atmospheric pressure at an altitude of 8000 ft. above sea level is: 10.87 lb. per square inch; hence:

$$W_2=2.7077\times\frac{60+10.87}{100+461}=0.3421 \text{ lb.}$$

CHAPTER III

THE COMPRESSION OF AIR IN AIR COMPRESSORS

25. The Air Cylinder of a Compressor.—Fig. 3 shows the air cylinder (A) of a reciprocating compressor, in which the air is compressed by a piston (B), whose rod (C) is connected to the piston of a steam engine or through a connecting rod and crank to a revolving shaft, the latter being driven by some form of prime mover.

The cylinder shown is that of a single-stage compressor, in which the air is compressed in one operation and in one cylinder, from initial to final pressure. In two- and multi-stage compressors the air is compressed gradually in succeeding cylinders, being cooled to in-take temperature while passing from one cylinder to the next one. (See Article 57.)

26. Water-jackets.—As shown in Fig. 3, the cylinder heads and usually the main body of the air cylinders are water-jacketed. The chief object of this is to prevent the cylinders from reaching a temperature which would vaporize the lubricating oil and thus cause rapid wear of piston and cylinder. Incidentally the air itself is cooled to some extent by the surrounding water, which means a gain in efficiency.

27. Inlet and Discharge Valves.—Modern compressors as a rule are double acting, that is, air is taken in, compressed, and discharged on the forward stroke as well as on the backward stroke of the piston. For this reason each of the cylinder heads carries one or more inlet valves a, a' through which the atmospheric air can enter the cylinder, and one or more discharge valves b, b' which open outward into closed ports g, h connected by a conduit c, which leads to a closed receiver R, whence the compressed air is conveyed to the place where it is proposed to use it. The inlet and discharge valves are either mechanically moved, resembling in their general form and operation the steam valves of a Corliss engine, or they are of the poppet type, being pressed upon their seats by a spring. In Fig. 3, which shows the section of the air cylinder of an air compressor, the inlet valves

a, *a′* are mechanically moved, the discharge valves *b*, *b′* are of the poppet type. For description of valves see Articles 144–153.

28. Analysis of Single-stage Compression.—At the beginning of the stroke all valves are closed. The piston moving from right to left, as shown in the figure, causes a partial vacuum behind it;

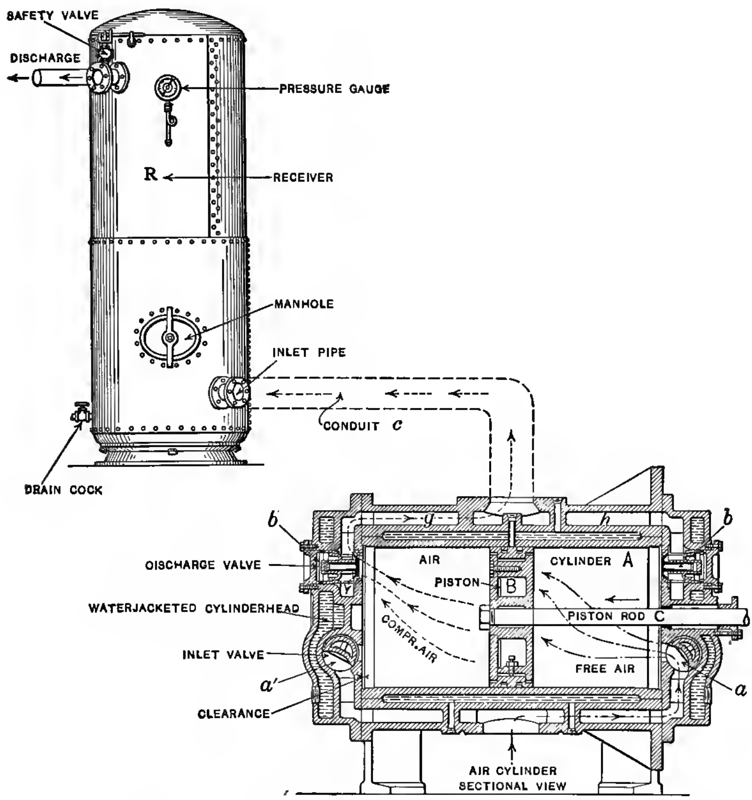


Fig. 3.—Diagram Illustrating Principle of Single-stage Compression.

the inlet valves open under atmospheric pressure (unless opened mechanically) and the outside or free air rushes into the cylinder behind the receding piston.

On the left-hand side of the piston we have at the beginning of the stroke a cylinder full of atmospheric or free air, which by the advancing piston is compressed into a steadily decreasing volume. The pressure of the air on this side of the piston is at

the same time steadily increasing until at a certain point of the stroke it reaches, or slightly surpasses, the receiver pressure. Beyond this point the increasing pressure causes the discharge valves to open, and to the end of the stroke the compressed air is delivered into the receiver under constant pressure.

If the inlet valves in the left-hand side of the cylinder are of the poppet type, they are kept closed during the forward stroke of the piston by the pressure of the air inside the cylinder, which is greater than the outside atmospheric pressure.

29. The Receiver pressure depends on the work which the compressed air is to perform. If, for instance, the air engines at the end of the pipe line require a pressure of 80 lb. gage, no air is to be drawn from the pipe line until the gage of the receiver near the compressor shows this pressure, plus the pressure required for transmission. After this, the supply of compressed air must keep pace with the demand. Should the demand exceed the supply, the pressure of the air would drop below 80 lb., thereby impairing the efficiency of the air engines. If on the other hand the supply at any time should exceed the demand, the pressure of the air in the receiver and in the pipe line would increase until it reaches the pressure for which the safety-valve of the receiver is set, when it will blow off through the latter.

As this means a waste of energy, compressors which furnish air for intermittent work are generally supplied with automatic regulating devices, such as are described and illustrated in Articles 157–160.

CHAPTER IV

THEORY OF AIR COMPRESSION

A. ISOTHERMAL COMPRESSION

30. According to Boyle's Law, at constant temperature the volume occupied by a given weight of air varies inversely as the absolute pressure:

$$\frac{V}{V_1}=\frac{P_1}{P}$$

whence

$$P_1=P\frac{V}{V_1}$$

in which V = the volume of a given weight of air at an absolute pressure P and a certain temperature.

V_1 = the volume of the same weight of air at the same temperature and at any absolute pressure P_1.

Take, for instance, 1 cu. ft. of free air at 60° Fahr., having an absolute pressure of one atmosphere or 14.7 lb. per square inch. Assume that this air is confined under the piston of a closed cylinder, and that driving the piston forward we reduce the volume occupied by the air to 1/2 cu. ft., at the same time maintaining its temperature at 60° Fahr., then the absolute pressure of the air would be $P_1=14.7\times\frac{1}{1/2}=29.4$ lb. per square inch, or twice what it was before.

If the volume were reduced to 1/3 cu. ft., its absolute pressure would become $3\times14.7=44.1$ lb. per square inch, or 29.4 lb. gage, always upon the condition that the temperature remains at 60° Fahr.

In other words, if the volume of air becomes 1/2, 1/3, 1/4, etc., times the original volume, its pressure becomes 2, 3, 4, etc., times the original pressure, always taking absolute pressures.

31. Graphical Illustrations of Isothermal Compression.—Let Fig. 4 represent the air cylinder of a compressor, 48 in. long with a piston moving in it in the direction of the arrow. Let

the cylinder be connected to a receiver in which the pressure is 73.5 lb. gage or six atmospheres per square inch.

Assume that the cylinder has been filled with atmospheric air during the suction stroke of the piston. By moving the piston 12 in. from the left to the right, the volume of the air is reduced to $\frac{48-12}{48}=\frac{3}{4}$ of the original volume and the pressure has increased to 4/3 times the atmospheric pressure, that is, to $4/3\times14.7=19.6$ lb. absolute or 4.9 lb. gage.

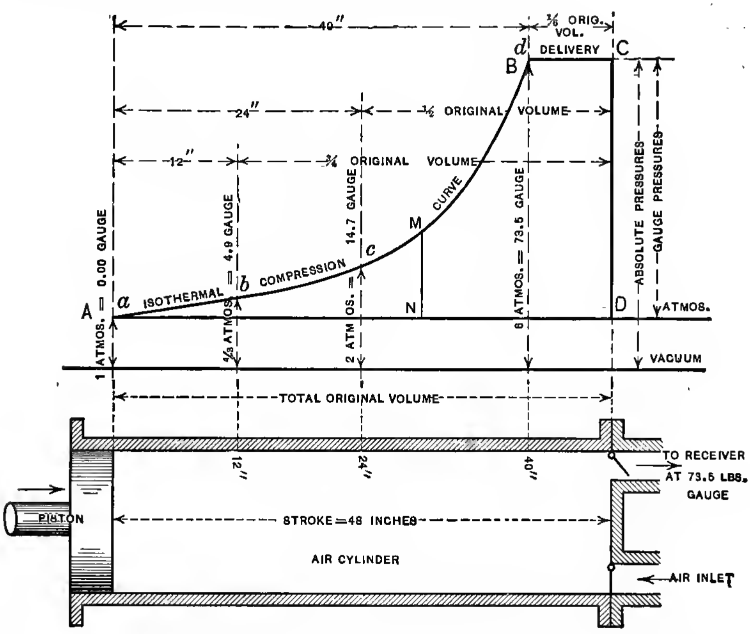


FIG. 4.—Diagram Illustrating Isothermal Compression and Delivery.

When the piston has advanced to a point 24 in. from its first position, the volume of air has been compressed to $\frac{48-24}{48}=\frac{1}{2}$ of the original volume and the pressure is now twice that of the atmosphere, that is, 29.4 lb. absolute or 14.7 lb. gage.

The same reasoning applies to other positions of the piston until the latter reaches a point 40 in. from the starting point. The air has now been reduced to $\frac{48-40}{48}=\frac{1}{6}$ of the original volume and the pressure has increased to six atmospheres, that is, 88.2

lb. absolute or 73.5 lb. gage. This being the pressure in the receiver, the discharge valves open and the remaining 8 in. of the stroke are completed by the piston against a constant pressure of 73.5 lb. in delivering the air into the receiver.

32. Construction of the Isothermal Compression Curve.—Draw a horizontal line, AD, which at a convenient scale represents 48 in. and mark on this line points at 12, 24, and 40 in. from its left end; then draw at those points lines perpendicular to AD.

On these lines, measure off, at any other convenient scale, the gage pressures corresponding to the stroke; this will give a succession of points a, b, c, d, and, if we join them by a continuous line, we get a curve $A–B$ which represents the variations of air pressure during the compression; that is, the gage pressure at any point M is measured by the line MN.

The curve AB is a hyperbola and is known as the curve of isothermal compression. Its equation is:

$$PV = \text{constant}$$

If we took any number of intermediate points between 40 and 48 in. of the stroke, the pressure would always be 73.5 lb. gage, consequently a line drawn connecting these points will be a straight line parallel to AD. It represents the period of delivery under constant pressure.

33. Work of Isothermal, Single-stage Compression and Delivery.—Work is the product of a force and the distance through which it acts in the direction of its application.

In the diagram, Fig. 5, let AB represent an isothermal compression curve

BC = the line of delivery
P_1 and P_2 = absolute initial and terminal pressures in pounds per square inch
L = length of stroke in feet.

The force acting on the body of air contained in the cylinder is the force applied to the piston by some external agent, such as steam, water, electricity, etc. The displacement of the point of application during one stroke of the piston is the length of the stroke (L).

The force applied to the piston must be equal (theoretically) to the resistance offered by the air inside the cylinder, that is, to

its pressure, which at any point of the stroke is proportional to the volume into which the air has been compressed.

During compression of the air from A to B the pressure increases from an absolute pressure P_1 to an absolute pressure P_2; during the remainder of the stroke from B to C the air which now occupies a volume V_2, represented by the distance BC, is delivered into the receiver at a constant absolute pressure P_2.

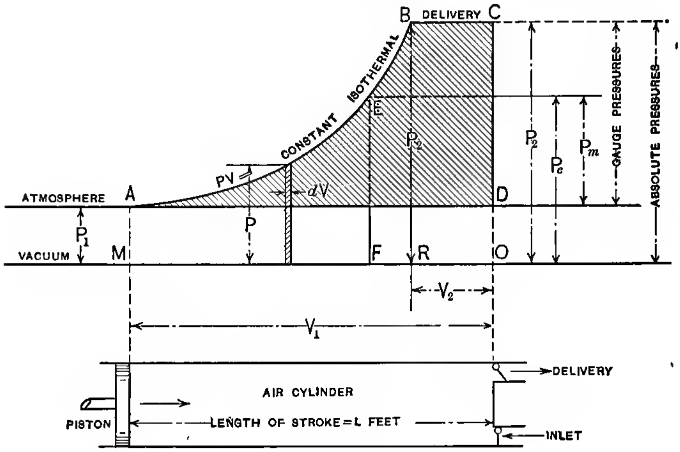


FIG. 5.

The average resistance of the air during the entire stroke is the mean pressure of the air against the piston. Its value in terms of absolute pressure in pounds per square inch is represented by a line EF, located somewhere between M and O.

Let A = area of piston in square feet
L = length of stroke in feet
P_1 = absolute initial pressure of in-take air in pounds per square inch
P_2 = absolute terminal pressure in pounds per square inch
P_e = mean absolute pressure in pounds per square inch
W = total work performed per stroke in foot-pounds
W_n = net work performed per stroke in foot-pounds
V_1 = volume of free air in cubic feet taken into the cylinder per stroke
V_2 = volume of air in cubic feet after being compressed to an absolute pressure P_2.

The total mean force acting on the piston during the entire stroke is 144 P_eA lb. and the total work performed per stroke is

$$W = 144\ P_eAL \text{ ft.-lb.}$$

But $$AL = V_1$$

hence $$W = 144 P_e V_1 \text{ ft.-lb.} \tag{1}$$

If in the diagram, Fig 5, we substitute for the length of the stroke (L) the volume V_1 in cubic feet of free air, taken into the cylinder per stroke, then the product P_eV_1 in equation (1) is equal to the numerical value of the area *MABCO*. This value is obtained by multiplying P_e, expressed in pounds per square inch, with V_1, expressed in cubic feet.

The total work done during one stroke of the piston is as follows:

w_1 = work of compressing a volume V_1 of free air from an absolute pressure P_1 to an absolute pressure P_2. This work is proportional to the area *MABR*.

w_3 = work of delivering the compressed air which now occupies a volume V_2 into the receiver under a constant absolute pressure P_2. This work is proportional to the area *BCOR*.

The sum of the two quantities w_1 and w_2 includes

w_3 = work of filling the cylinder behind the advancing piston with a new volume of free air which is to be compressed during the return stroke of the piston. This work is proportional to the area *MADO*.

The work w_3, however, is not performed by energy supplied by the compressor, but by the pressure of the in-take (atmospheric) air. Measured in foot-pounds it is

$$w_3 = 144\ P_1V_1$$

in which the product P_1V_1 is equal to the numerical value of the area *MADO*.

34. The net work W_n in foot-pounds performed by the compressor during one stroke of the piston is, therefore:

$$W_n = 144 \times (\text{area } MABCO \text{ minus area } MADO)$$
$$= 144 \times (\text{shaded area } ABCD)$$

But area $ABCD =$ area $MABR$
plus area $BCOR$
minus area $ADOM$

$$\text{Area } MABR = \int_{V_2}^{V_1} P\,dV$$

For isothermal compression $PV = P_1V_1$ or $P = P_1\frac{V_1}{V}$

Hence
$$\text{Area } MABR = P_1V_1\int_{V_2}^{V_1}\frac{dV}{V}$$

$$= P_1V_1 \text{ Naperian log } \frac{V_1}{V_2}$$

and since
$$\frac{V_1}{V_2} = \frac{P_2}{P_1}$$

$$\text{Area } MABR = P_1V_1 \text{ Naperian log } \frac{P_2}{P_1}$$
$$\text{Area } BCOR = P_2V_2$$
$$\text{Area } ADOM = P_1V_1$$

Therefore, $W_n = 144\left(P_1V_1 \text{ Naperian log } \frac{P_2}{P_1} + P_2V_2 - P_1V_1\right)$
and since under isothermal conditions $P_2V_2 = P_1V_1$.

35. Net Work of Isothermal Compression and Delivery per Stroke:

$$W_n = 144\,P_1V_1 \log_e \frac{P_2}{P_1} \text{ ft.-lb.}$$

in which P_1 = initial absolute pressure in pounds per square inch.
P_2 = final absolute pressure in pounds per square inch.
V_1 = volume of free air in cubic feet taken into the cylinder per stroke.
$\log_e$ = Naperian log = 2.302585 times common log.

36. Mean Gage—Pressure, Isothermal Compression and Delivery:

Let P_m = mean gage pressure in pounds per square inch.
A = area of piston in square feet.
L = length of stroke in feet.

V_1 = volume of air in cubic feet taken into the cylinder per stroke.
W_n = net work per stroke in foot-pounds.

Then $W_n = 144\ P_m AL$ and since $AL = V_1$

$$W_n = 144\ P_m V_1 \text{ whence } P_m = \frac{W_n}{144\ V_1}$$

or $$P_m = P_1 \log_e \frac{P_2}{P_1} \text{ lb. per square inch.}$$

Column 6 of Table III gives mean gage pressures for isothermal compression and delivery and for various terminal gage pressures.

37. Theoretical Horse-power, Isothermal Single-stage Compression and Delivery.—The theoretical horse-power required to compress isothermally in one stage a volume V_1 of free air per minute from an absolute pressure P_1 to an absolute pressure P_2 and deliver the compressed air into the receiver under constant pressure is found from the general formula:

$$\text{Horse-power} = \frac{PLAN}{33{,}000}$$

in which P = mean gage pressure in pounds per square inch.
L = length of stroke in feet.
A = area of piston in square inches.
N = number of strokes per minute.

If we now let V_1 designate the volume of free air in cubic feet taken into the cylinder per minute, we have:

$$V_1 = \frac{A}{144} LN$$

from which $$LAN = 144\ V_1$$

Substituting this value and the value P_m for P in our formula, we get

$$\text{Horse-power} = \frac{144\ P_1 V_1}{33{,}000} \log_e \frac{P_2}{P_1}$$

in which V_1 = volume of free air in cubic feet taken into the cylinder per minute.
P_1 = initial absolute pressure in pounds per square inch.
P_2 = terminal absolute pressure in pounds per square inch.
$\log_e$ = Naperian log = 2.302585 times common log.

Column 3 of Table V gives the theoretical horse-power required to compress 1 cu. ft. of free air per minute isothermally in one stage to various gage pressures and deliver it at that pressure into the receiver.

38. Isothermal compression in actual practice is impossible of attainment. It is only approached in slow-speed compressors, where the air is in contact with the water-jackets for a longer time than in normal speed machines.

In actual practice the compression curve as obtained from indicator diagrams falls closer to the adiabatic than to the isothermal curve. For this reason the formulas for adiabatic compression are generally used in practical compressor computations.

39. Isothermal Expansion.—If compressed air could be expanded isothermally down to atmospheric pressure in an air engine, the theoretical work performed would be the same as the work required for isothermal compression and delivery. Hence, work of isothermal expansion:

$$\text{Horse-power} = \frac{144 \ P_1 V_1}{33{,}000} \log_e \frac{P_2}{P_1} \qquad (1)$$

in which P_1 = absolute pressure of exhaust air in pounds per square inch.

= atmospheric pressure.

V_1 = volume in cubic feet per minute, which the volume V_2 of compressed air, admitted into the air engine per minute, would occupy after expansion to initial pressure.

P_2 = absolute pressure in pounds per square inch of the air admitted into the air engine.

$\log_e$ = Naperian log = 2.302585 times common log.

In expansion work we usually know the volume V_2 of compressed air taken into the cylinder of an air engine per unit of time, and since under isothermal conditions

$$P_1 V_1 = P_2 V_2$$

We can also write:

Theoretical horse-power isothermal expansion:

$$\text{Horse-power} = \frac{144 \ P_2 V_2}{33{,}000} \log_e \frac{P_2}{P_1} \qquad (2)$$

in which P_2 = absolute pressure of air taken into cylinder in pounds per square inch.

P_1 = exhaust (atmospheric) pressure.

V_2 = volume of compressed air in cubic feet taken into the cylinder per minute.

At the present stage of the art, isothermal expansion is impossible of attainment in actual practice. The formulas introduced under this article, however, are useful for comparison and for estimating efficiencies of compressores, pipe lines and air engines.

B. ADIABATIC COMPRESSION OF AIR

Theory

40. We have seen that if air is compressed, heat is generated.

In adiabatic compression this heat is allowed to accumulate unchecked during the period of compression. As a consequence, when a certain pressure is reached, the corresponding volume of air will be greater on account of this heat than the volume which the air would occupy if the compression up to that same pressure had been isothermal. When the volume is reduced to one-half, the pressure is not only double as in isothermal compression, but more than double because of the heat, generated during compression, being still in the air.

Again, when the pressure has been doubled, the volume will not be one-half, but will be more than one-half, owing to the expansion due to heat which has remained in the air.

Since the pressure rises faster than the volume diminishes, $\frac{P}{P_1}$ is no longer equal to but is greater than $\frac{V_1}{V}$. To form an equation, the value of $\frac{V_1}{V}$ must be increased. This is done by introducing an exponent "n" which raises the value of $\frac{V_1}{V}$ to a power whose index has been found to be the ratio between the specific heat of air at constant pressure, and the specific heat at constant volume, expressed either in heat units (B.T.U.'s) or in foot-pounds.

$$n = \frac{C_p}{C_v} = \frac{0.2375}{0.1689} = \frac{K_p}{K_v} = \frac{184.8}{131.6} = 1.406 \qquad (1)$$

This gives for the general equation of the adiabatic compression or expansion curve:

$$PV^n = P_1V_1^n \tag{2}$$

and, since the exponent n takes care of the changes in temperature, due to adiabatic compression or expansion, we have from analogy with deductions made under Article 16:

$$PV^n = P_1V_1^n = \text{constant} \tag{3}$$

The synthetical method by which "n" is found to equal $\frac{K_p}{K_v}$ is shown in Article 117a.

The value of n varies slightly with the variation in the specific heats of air, due to the presence of moisture, as pointed out under Article 8. Throughout this treatise, the value

$$n = 1.406$$

will be used.

RELATION BETWEEN TEMPERATURE, VOLUME AND PRESSURE IN ADIABATIC SINGLE STAGE COMPRESSION OR EXPANSION OF AIR

41. The relation between temperature, pressure, and volume of air at the beginning and at the end of adiabatic single stage compression or expansion can be deduced from Charles' and Boyle's Laws as follows: according to these laws (see Article 20, equation (3)).

$$P_1V_1 = PV\frac{T_1}{T} \tag{1}$$

whence

$$\frac{P_1}{P} = \frac{VT_1}{V_1T} \tag{2}$$

For adiabatic compression

$$\frac{P}{P_1} = \left(\frac{V_1}{V}\right)^n \tag{3}$$

or

$$\frac{P_1}{P} = \left(\frac{V}{V_1}\right)^n \tag{4}$$

combining equations (2) and (4) $\left(\frac{V}{V_1}\right)^n = \frac{VT_1}{V_1T}$

whence $$\left(\frac{V}{V_1}\right)^{n-1}=\frac{T_1}{T} \tag{5}$$

and $$\frac{V}{V_1}=\left(\frac{T_1}{T}\right)^{\frac{1}{n-1}}$$

or $$\frac{V_1}{V}=\left(\frac{T}{T_1}\right)^{\frac{1}{n-1}} \tag{6}$$

from equation (3) $$\frac{V_1}{V}=\left(\frac{P}{P_1}\right)^{\frac{1}{n}} \tag{7}$$

or $$\frac{V}{V_1}=\left(\frac{P_1}{P}\right)^{\frac{1}{n}}$$

whence $$\left(\frac{V}{V_1}\right)^{n-1}=\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}}$$

combining with equation (5) $$\frac{T_1}{T}=\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} \tag{8}$$

and $$\left(\frac{T_1}{T}\right)^{\frac{n}{n-1}}=\frac{P_1}{P} \tag{9}$$

This gives the following relations between volume, pressure, and temperature in adiabatic single stage compression or expansion: From (5) absolute temperature in terms of volumes

$$T_1=T\left(\frac{V}{V_1}\right)^{n-1} \tag{10}$$

From (8) absolute temperature in terms of absolute pressures

$$T_1=T\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} \tag{11}$$

From (6) volume in terms of absolute temperatures

$$V_1=V\left(\frac{T}{T_1}\right)^{\frac{1}{n-1}} \tag{12}$$

From (7) volume in terms of absolute pressures

$$V_1 = V\left(\frac{P}{P_1}\right)^{\frac{1}{n}} \tag{13}$$

From (9) absolute pressure in terms of absolute temperatures

$$P_1 = P\left(\frac{T_1}{T}\right)^{\frac{n}{n-1}} \tag{14}$$

From (4) absolute pressure in terms of volumes

$$P_1 = P\left(\frac{V}{V_1}\right)^{n} \tag{15}$$

in which V = volume corresponding to an absolute pressure P and an absolute temperature T.

V_1 = volume corresponding to an absolute pressure P_1 and an absolute temperature T_1 and *vice versa.*

n = exponent of adiabatic compression (=1.406).

41a. Law of Thermodynamics, Applied to Adiabatic Compression and Expansion of Air.—According to the law quoted under Article 6 heat and work are mutually convertible. In adiabatic compression of air all the work of compression is converted into heat (see Article 117), and the temperature of the air is increased correspondingly.

In compliance with the law referred to, a volume of compressed and therefore heated air, if allowed to expand adiabatically to initial pressure, against an external resistance, would perform work by converting back into mechanical energy all the heat received during compression. Theoretically, the amount of work performed will be equal to the work of compression. The temperature of the expanded air will be the same as before compression.

In compressed-air installations, practically all of the compression heat is abstracted from the air by water-cooling and radiation, previous to expansion. In this case the compressed air is still capable of doing expansive work as before, by converting heat into mechanical energy. But the capacity for doing work will be less than in the first case, due to the loss of the compression heat which is equivalent to a loss of energy.

It is obvious that in the second case the heat required to do work must come from some source other than the compression work. As a matter of fact, it is heat which was contained in the air before compression. That the expansive work consumes some of this heat is manifested by the cold created around the cylinders of an engine using air expansively.

The actual temperatures of the compressed and of the expanded air under various conditions may be determined by applying the laws and formulas given in preceding articles.

Example.—Let a volume $V=10$ cu. ft. of free air be adiabatically compressed in one stage from atmospheric pressure ($P=14.7$ lb. absolute) to 80 lb. gage ($P_1=94.7$ lb. absolute); the initial temperature of the air being 60° Fahr. ($T=521$ degrees absolute).

From equation (11), Article 41, we deduce the absolute temperature (T_1) of the air after adiabatic compression:

$$T_1=T\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} = 521\left(\frac{94.7}{14.7}\right)^{0.29} = 894.25^\circ \text{ absolute.} \quad = 433.25^\circ \text{ Fahr.}$$

The volume V_1 into which the air has been compressed under adiabatic conditions, we find from equation (13), Article 41:

$$_1V=V\left(\frac{P}{P_1}\right)^{\frac{1}{n}} = 10\left(\frac{14.7}{94.7}\right)^{0.71} = 2.664 \text{ cu. ft.}$$

If we cool this volume V_1 of compressed and heated air to initial temperature of 60° Fahr. ($T=521$ degrees absolute), the effect, according to Article 5, will be a decrease of pressure under constant volume. Calling the new pressure P_3, we find same by applying the law stated under Article 19:

$$\frac{P_3}{P_1}=\frac{T}{T_1}$$

whence $$P_3=P_1\frac{T}{T_1} = 94.7\ \frac{521}{894.25} = 55.174 \text{ lb. absolute.}$$

The volume of the cooled air has, of course, remained the same, viz.: 2.664 cu. ft.

If the air, occupying a volume V_1 at an absolute pressure of 55.174 lb. and a temperature of 60° Fahr., were allowed to expand adiabatically down to atmospheric pressure, the temperature of the expanded air, according to equation (11), Article 41, would be:

$$T_2=T\left(\frac{P}{P_3}\right)^{\frac{n-1}{n}} = 521\left(\frac{14.7}{55.174}\right)^{0.29} = 355.06 \text{ degrees absolute.} \quad = -105.94^\circ \text{ Fahr.}$$

and from equation (13), Article 41, the volume V_2 of the expanded air would be:

$$V_2 = V_1\left(\frac{P_3}{P}\right)^{\frac{1}{n}} = 2.664\left(\frac{55.174}{14.7}\right)^{0.71} = 6.8143 \text{ cu. ft.}$$

This is the same quantity (weight) of atmospheric air we started out to compress, but, being so much colder (−105.94° Fahr.), occupies a smaller volume than the original volume of 10 cu. ft. If we now heat this cold exhaust air to initial temperature (60° Fahr.), under constant pressure, we should get our original volume. Applying the law stated under Article 18, we have:

$$\frac{V}{V_2} = \frac{T}{T_2}$$

whence $$V = V_2\frac{T}{T_2} = 6.8143\ \frac{60+461}{355.06} = 10.00 \text{ cu. ft.}$$

42. Graphical Illustration of Adiabatic Compression.—Assume a cylinder (Fig. 6) whose stroke is 48 in. and which

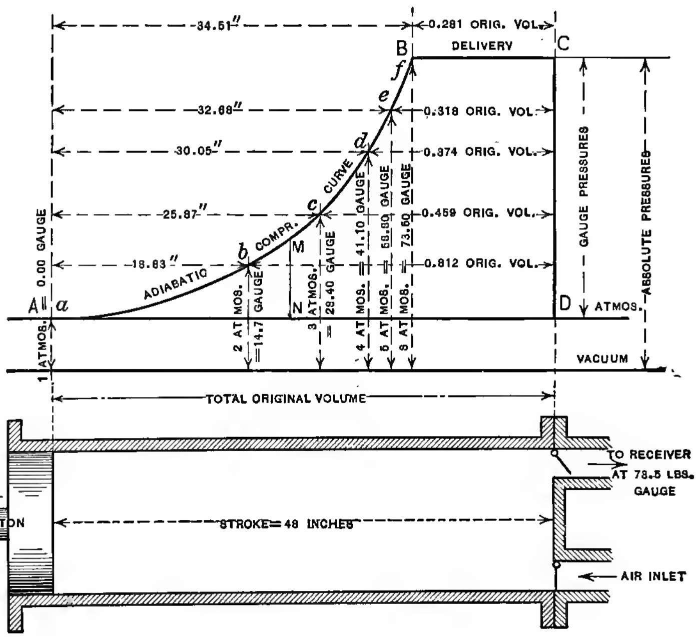


FIG. 6.—Diagram Illustrating Adiabatic Compression and Delivery.

is filled with air at atmospheric pressure at a temperature of 60° Fahr. and having a piston moving in it in the direction of

the arrow. Let the cylinder be connected with a receiver in which the pressure is six atmospheres or 73.5 lb. gage.

If we move the piston from left to right and compress the air in the cylinder to two atmospheres, the volume will not be one-half the original volume (as in isothermal compression) but will be greater than one-half.

From column 3 of Table IV we find that the volume is 0.612 times the original volume, hence the piston will be at a point $48-(0.612 \times 48)=18.63$ in. from the left end of the cylinder. Thus we find the position of the piston at a pressure

of 2 atmospheres to be at $48-(0.612 \times 48)=18.63$ in.
of 3 atmospheres to be at $48-(0.459 \times 48)=25.97$ in.
of 4 atmospheres to be at $48-(0.374 \times 48)=30.05$ in.
of 5 atmospheres to be at $48-(0.319 \times 48)=32.69$ in.
of 6 atmospheres to be at $48-(0.281 \times 48)=34.51$ in.

from the left-hand end of the cylinder.

43. Construction of the Adiabatic Compression Curve.—Draw a horizontal line AD which, at a convenient scale represents 48 in., and mark on this line points at 18.63, 25.97, 30.05, 32.69, and 34.51 in. from the left end; then draw at those points lines perpendicular to AD.

On these lines measure off, at any scale, the gage pressures corresponding to the stroke; this will give a succession of points a, b, c, d, e, f, and if we join these by a continuous line we get a curve AB which represents the variations of air pressure during compression, that is, the gage pressure at any point M is measured by the line MN.

The curve AB is known as the curve of adiabatic compression.

If we took any number of intermediate points between 34.51 and 48 in. of the stroke, the pressure would always be 73.5 lb. gage, and consequently a line connecting these points will be a straight line BC parallel to AD. It represents the period of delivery under constant pressure.

WORK OF ADIABATIC SINGLE-STAGE COMPRESSION AND DELIVERY

44. The net work in foot-pounds performed during one complete stroke of the piston in compressing adiabatically a volume V_1 of free air from an absolute pressure P_1 to an absolute pressure

P_2 and in delivering the compressed air which now occupies a volume V_2, under constant pressure P_2 into the receiver, is obtained in the identical manner as has been shown for isothermal compression.

Referring to Fig. 7, net work performed by the compressor under the conditions named is $W_n = 144 \times$ (shaded area $ABCD$) ft. lb.

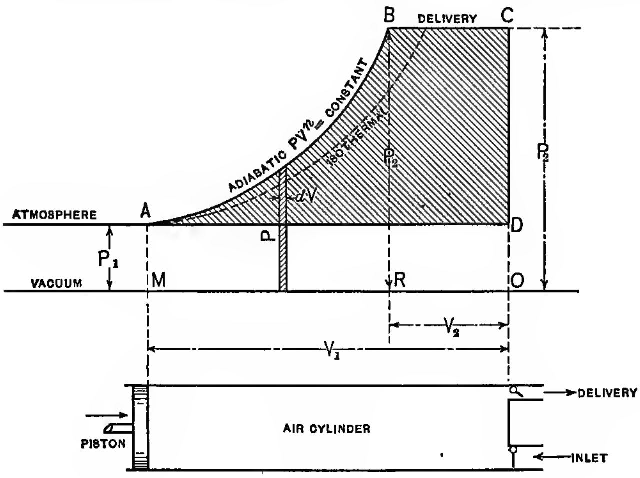


FIG. 7.

But area $ABCD$ = area $MABR$
plus area $BCOR$
minus area $ADOM$

$$\text{Area } MABR = \int_{V_2}^{V_1} P\,dV$$

For adiabatic compression

$$PV^n = P_1 V_1^n \text{ or } P = \frac{P_1 V_1^n}{V^n}$$

Hence
$$\text{area } MABR = \int_{V_2}^{V_1} \frac{P_1 V_1^n}{V^n}\,dV$$

$$= P_1 V_1^n \int_{V_2}^{V_1} \frac{dV}{V^n}$$

$$=P_1V_1^n\int_{V_2}^{V_1}V^{-n}dV$$

$$=P_1V_1^n\left(\frac{V_1^{1-n}-V_2^{1-n}}{1-n}\right)$$

$$=\frac{P_1V_1^nV_1^{1-n}-P_1V_1^nV_2^{1-n}}{1-n}$$

And since $P_1V_1^n=P_2V_2^n$

We can write area $MABR=\dfrac{P_1V_1^nV_1^{1-n}-P_2V_2^nV_2^{1-n}}{1-n}$

or, area $MABR=\dfrac{P_2V_2-P_1V_1}{n-1}$

Area $BCOR=P_2V_2$
Area $ADOM=P_1V_1$

Therefore area $ABCD=\dfrac{P_2V_2-P_1V_1}{n-1}+P_2V_2-P_1V_1$

$$=\frac{P_2V_2-P_1V_1+nP_2V_2-nP_1V_1-P_2V_2+P_1V_1}{n-1}$$

$$=\frac{n}{n-1}P_1V_1\left(\frac{P_2V_2}{P_1V_1}-1\right)$$

But $\dfrac{V_2}{V_1}=\left(\dfrac{P_1}{P_2}\right)^{\frac{1}{n}}$

Substituting we get

Area $ABCD=\dfrac{n}{n-1}P_1V_1\left[\left(\dfrac{P_2}{P_1}\right)^{\frac{n-1}{n}}-1\right]$ and

45. Net work per stroke $W_n=\dfrac{144\,n}{n-1}P_1V_1\left[\left(\dfrac{P_2}{P_1}\right)^{\frac{n-1}{n}}-1\right]$ ft.-lb.

in which P_1 = initial absolute pressure in pounds per square inch.
P_2 = final absolute pressure in pounds per square inch.
V_1 = volume of free air in cubic feet taken into the cylinder per stroke.
n = exponent adiabatic compression (1.406)

3

46. Mean Gage Pressure, Adiabatic Single-stage Compression and Delivery.—The mean gage pressure in pounds per square inch in single-stage adiabatic compression and delivery is deduced in the same manner as shown for isothermal compression. It is

$$P_m = \frac{W_n}{144\,V_1} \text{ or}$$

$$P_m = \frac{n}{n-1}P_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ lb. per square inch.}$$

Column 7 of Table III gives the theoretical mean gage pressure in pounds per square inch for adiabatic single stage compression and delivery.

47. Theoretical Horse-power, Adiabatic Single-stage Compression and Delivery.—The theoretical horse-power required to compress adiabatically a volume V_1 of free air in one stage from an absolute pressure P_1 to an absolute pressure P_2 and to deliver the compressed air into the receiver at a constant pressure P_2 is found from the general formula

$$\text{Horse-power} = \frac{PLAN}{33{,}000}$$

in which P = mean gage pressure in pounds per square inch.
L = length of stroke in feet.
A = area of piston in square inches.
N = number of strokes per minute.

If by V_1 we designate the volume of free air in cubic feet taken into the cylinder per minute, we have

$$V_1 = \frac{A}{144}LN$$

whence $LAN = 144V_1$. Substituting this value and the value P_m for P in our formula we get

$$\text{Horse-power} = \frac{144\,P_1V_1n}{33{,}000(n-1)}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad (1)$$

in which V_1 = volume of free air in cubic feet to be compressed and delivered per minute.
P_1 = initial absolute pressure in pounds per square inch.
P_2 = terminal pressure in pounds per square inch.
n = exponent of adiabatic compression (1.406)

A glance at Fig. 7 shows that the work of adiabatic compression and delivery is greater than that of isothermal compression and delivery. In actual practice the indicator card from an air cylinder of a compressor running at ordinary speed, shows a compression line approaching the adiabatic curve much more closely than the isothermal; so closely that in making computations it is usually assumed that the compression has been adiabatic.

Column 4 of Table V gives the theoretical horse-power required for adiabatic single-stage compression and delivery of 1 cu. ft. of free air per minute ($V_1 = 1.00$) at sea level.

48. Theoretical horse-power, single-stage adiabatic compression and delivery, expressed in terms of absolute temperatures and weight of the volume of air to be compressed and delivered per minute:

According to Article 20, equation (5):

$$P_1 V_1 = R T_1 \tag{1}$$

in which P_1 = absolute pressure of air in pounds per square foot.
V_1 = volume in cubic feet of 1 lb. of air at an absolute pressure P_1 and an absolute temperature T_1.

From equation (7), Article 20, we have:

$$R = (K_p - K_v) = 184.8 - 131.6 = 53.2.$$

From Article 40, equation (1), we have:

$$\frac{K_p}{K_v} = n$$

whence

$$K_v = \frac{K_p}{n}$$

whence

$$R = K_p - \frac{K_p}{n} = \frac{n-1}{n} K_p \tag{2}$$

From equation (9) in Article 41 we deduce:

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{n}{n-1}}$$

In substituting in the horse-power formula, Article 47, the value for $P_1 V_1$ as given in equation (1) of this article, it must be remembered that in equation (1) the pressure P_1 is expressed in *pounds per square foot* and the volume V_1 is the volume in cubic feet of 1 lb. of free air, whereas in the horse-power formula, Article 47,

P_1 is expressed in *pounds per square inch* and V_1 is the volume of free air in cubic feet to be compressed per minute. Therefore if we wish to compress adiabatically and deliver w pounds of free air per minute, the formula becomes

$$\text{Horse-power} = w\frac{nRT_1}{(n-1)33{,}000}\left[\left(\frac{T_2}{T_1}\right)^{\frac{n}{n-1}\times\frac{n-1}{n}} - 1\right]$$

Introducing value of R from (2)

$$\text{Horse-power} = w\frac{n(n-1)K_pT_1}{n(n-1)33{,}000}\left[\frac{T_2 - T_1}{T_1}\right]$$

$$= w\frac{K_p}{33{,}000}[T_2 - T_1]$$

Introducing the value for $K_p = 184.8$ we get

$$\text{Theoretical horse-power} = 0.0056\, w\, [T_2 - T_1] \tag{3}$$

in which w = weight of the number of cubic feet of free air which are to be compressed and delivered per minute.

T_2 = final absolute temperature of compressed air.

T_1 = initial, absolute temperature of free air.

Example.—Find theoretical horse-power required at sea level to compress and deliver 100 cu. ft. of free air per minute, having an initial temperature of 60° Fahr., the final pressure to be 85 lb. gage.

From Article 23 we find the weight of 100 cu. ft. of atmospheric air at 60° Fahr.

$$w = 100\frac{39.804}{60+461} = 7.64 \text{ lb.}$$

From equation (11), Article 41, we find the absolute temperature of the air after compression:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$= (60+461)\left(\frac{85+14.7}{14.7}\right)^{0.29} = 908°$$

$$T_1 = (60+461) = 521°$$

$$T_2 - T_1 = 387°$$

Theoretical horse-power $= 0.0056 \times 7.64 \times 387 = 16.55$

RELATION BETWEEN FINAL PRESSURE OF A GIVEN QUANTITY OF AIR AND THE POWER REQUIRED TO COMPRESS TO THAT PRESSURE

49. The formulas for the horse-power required to compress and deliver a certain volume of free air, show that this power is not directly proportional to the final pressure. For in the quotient $\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$ the pressures are absolute pressures, that is, gage plus atmospheric pressures.

Doubling the gage pressure would not double the expression $\frac{P_2}{P_1}$ and therefore would not double the horse-power required to produce that pressure in the air. *To illustrate:* For a final gage pressure of 80 lb.

$$\text{The quotient } \frac{P_2}{P_1} \text{ is } \frac{80+14.7}{14.7}=6.442$$

$$\text{for 160 lb. } \frac{P_2}{P_1} \text{ is } \frac{160+14.7}{14.7}=11.88$$

$$\text{for 240 lb. } \frac{P_2}{P_1} \text{ is } \frac{240+14.7}{14.7}=17.33$$

Referring to column 7 of Table V, to compress 100 cu. ft. of free air per minute in two stages to 60 lb. gage requires 12.10 h.p. (theoretically). To compress to 180 lb., which is three times as much, only requires 20.8 h.p. which is less than twice the horse-power required for compression to 60 lb.

This points to conditions pertaining to compressed air which are advantageous in the transmission and final use in air engines as pointed out under Article 103.

50. Modified Power Values for Practical Air Compression Problems.—In the preceding theoretical formulas no allowance has been made for clearance, the heating of the intake air in passing through the valves, and the friction of the compressor.

The effect of the first two items on the consumption of power is negligible in good compressors. The additional energy required to overcome frictional resistance will amount in well-designed compressors to from 7 to 15 per cent. of the theoretical horse-power, depending to a great extent on the care that is taken with the machine.

For practical compressor computations an addition of 15 per cent. is usually made to the horse-power derived from the theoretical formulas.

CHAPTER V

CLEARANCE, VOLUMETRIC-EFFICIENCY, CAPACITY, SPEED, MECHANICAL-EFFICIENCY OF COMPRESSORS

51. Clearance.—This is the space enclosed between the piston and the cylinder head at the end of the stroke. See Fig. 3.

The clearance space, though generally a source of loss, is necessary for practical reasons: first, to avoid danger to the cylinder heads by allowing space for the water that may accumulate in the cylinders, and second, to provide passage sufficiently large for ready admission and delivery of the air.

It is evident that the clearance volume depends upon the area of the piston. In short-stroke cylinders of large diameter the clearance volume is a large proportion of the *Piston Displacement.* The latter is the actual volume swept through by the piston in one stroke.

Clearance is usually expressed as a ratio between clearance volume and cylinder volume. For cylinders of same diameter but different length of stroke, the ratio is larger in the short-stroke cylinder. In large, up-to-date compressors it varies from 1 to 2 per cent. It is much more in very small, short-stroke machines.

If the volume swept through by the piston in one stroke is 1000 cu. in. and the clearance volume is 20 cu. in. the compressor has 2 per cent. clearance.

52. Losses Due to Clearance.—If the discharge pressure is 75 lb. gage or 89.7 lb. absolute and the initial pressure is atmospheric pressure at sea level, that is, 14.7 lb. absolute, the air remaining in the 20 cu. in. clearance space will expand on the return stroke of the piston to about six times the clearance volume, or to 120 cu. in. and will, therefore, take up an additional 100 cu. in. from the in-take cylinder. That is, in a cylinder of 1000 cu. in. piston displacement the piston must travel back 10 per cent. of the return stroke before the clearance air has expanded to atmospheric pressure and before the atmospheric air is allowed to flow into the cylinder.

The actual room for the admission of new free air is therefore only $1000-100=900$ cu. in., or, as commonly stated, *the volumetric efficiency* of the compressor is 90 per cent.

Theoretically, the clearance loss, so called, is one of volumetric efficiency only and not of power. For although this air required work in compressing it to receiver pressure, in expanding it helps to compress the air on the other side of the piston. The loss of power due to loss of heat during expansion of the clearance air usually is a negligible quantity.

In practice, a loss of power is caused by clearance, due to the fact that, in order to deliver a definite amount of air, a larger compressor, consuming more power, is required.

The loss of volumetric efficiency due to clearance is less for two-stage than for single-stage compression, because for any given capacity the low-pressure cylinder of the two-stage machine is practically of the same size and has the same percentage of clearance as the cylinder of a single-stage machine. But the terminal pressure in the low-pressure cylinder of the two-stage machine is much lower, hence the expansion of the clearance air back into the cylinder volume is much less, and as a consequence the volumetric efficiency is higher. (See Chapter VI on Compound Compression.)

53. The volumetric efficiency of a compressor is the ratio of the volume of free air actually admitted and compressed in the in-take cylinder to the piston displacement.

The diagram in Fig. 8 represents an ideal air card, in which

GA is the admission line.
AB is the compression line.
BC is the delivery line.
CG is the expansion line.

This diagram shows graphically the loss in volumetric efficiency due to clearance and also that due to imperfections in the admission of free air. In order to cause the outside air to flow into the cylinder, the pressure in the latter must be less than the atmospheric pressure; the admission line will therefore always fall more or less below the atmospheric line as shown exaggerated on the diagram. If the in-take areas are restricted, the drop in pressure may become considerable. The clearance volume is represented by the lines $EF = CD$.

FG is the extra volume occupied by the clearance air after expansion and if there were no other losses, the volumetric efficiency would be represented by the line *AG*. But on the forward stroke the piston must travel a distance *AR* before the atmospheric line is reached and before actual compression begins. Hence the actual volumetric efficiency in the case illustrated by the diagram is represented by the line *RG*.

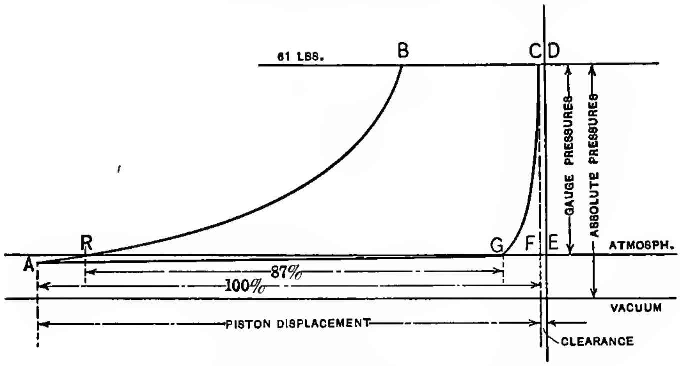


FIG. 8.—Air Card of a Single-stage Compressor.

High-class large compressors have a volumetric efficiency of over 90 per cent. In small single-stage compressors with insufficient water cooling, restricted inlet areas, and leaking pistons, the volumetric efficiency will be found much below the above figure.

It will be observed from the above that, in making calculations for a compressor plant to furnish a certain amount of free air per minute, it is quite important to make due allowance for volumetric efficiency. In installing a compressor plant, it is a wise precaution to have the builder guarantee a definite minimum volumetric efficiency.

54. Capacity.—Compressor builders frequently call the free-air capacity of their machines the volume swept through by the piston, without making any deductions, that is, if the area of the piston is 2 sq. ft. and the latter travels 500 ft. per minute, the capacity is called 1000 cu. ft. per minute.

Now, we have seen that if the clearance of a compressor, compressing to 75 lb. gage, is 2 per cent. the actual capacity is only 900 cu. ft. per minute and if 1000 cu. ft. capacity is wanted of the same compressor it must be speeded up to 555 ft. per minute.

Clearance is one of the factors affecting capacity. Since the volume which the expanded clearance air occupies, increases as the pressure increases, it follows that the loss in capacity by clearance is directly proportional to the pressure.

Another factor affecting compressor capacity is the condition of the air taken into the cylinder, which in common practice of rating capacities is assumed to be at atmospheric pressure and at no higher temperature than the outside source of supply. Such ideal conditions never exist. Even with unobstructed inlet passages air will not flow into the cylinder without some difference in pressure to force it in; hence the air taken into the cylinder always has a pressure slightly below that of the outside air. Then again, the entering air, coming in contact with the cylinder walls which have been highly heated during the compression in the preceding stroke, is heated to a temperature higher than outside temperature, thereby decreasing in density. As a consequence, at each stroke the compressor takes in a volume of air which weighs less than the same volume would weigh had it remained at outside temperature and pressure.

The temperature of a given volume of air does not affect the power required to compress and deliver that volume, it merely expands or contracts the product.

Table VII shows the effect of initial in-take temperature upon the efficiency and capacity of a compressor.

55. Speed of Compressors.—From what has been said concerning the capacity of a compressor, it might be assumed that this capacity could be indefinitely increased by increasing the piston speed. This is true to a certain degree. The question is, where is the limit? The only general answer that can be given is, when it does no longer pay, in dollars and cents.

Since the cost of compressing air must depend to a large extent on industrial conditions, the price of labor, fuel, supplies, etc., in the locality where the air is to be used, the ultimate speed of an air compressor is usually the result of a compromise between first cost, operating cost, and efficiency.

There are other factors which make speeds beyond a certain limit objectionable. It has been shown that in order to cause the air to flow into the in-take cylinder, there must be a difference between the pressures inside and outside of the cylinder. The outside pressure being normal atmospheric pressure, the pressure within the cylinder on the in-take-stroke must, therefore, always

be something less than this. Increased piston speed demands an increase of velocity at which the air must flow into the cylinder, and this in its turn demands a greater difference between the inside and outside pressure, that is, a reduced pressure in the in-take cylinder.

From the diagram in Fig. 8 it will be seen that a large drop in pressure of the in-take air below atmospheric pressure means a lengthening of the distance AR, hence a decrease of volumetric efficiency.

In following up this reasoning it becomes evident that a large increase of speed will usually give but a small increase in delivery, besides causing a rapid wear and frequent breakage of the working parts of the machine, to say nothing of the increased friction which in its turn reduces the mechanical efficiency of the compressor.

Increased speed, furthermore, means increase in temperature because the air is rushed through the cylinder at a greater velocity and does not come in contact with the water-jackets long enough to be cooled to any great extent. This rise of temperature increases the difficulty of lubrication and, if carried far enough, it may reach the ignition point of combustible substances in the cylinders, causing an explosion with all the expensive delays and repairs incident thereto.

The catalogues of compressor builders show piston speeds of compressors all the way from 150 ft. per minute for small, single-stage, 6-in. stroke machines with a rated capacity of 30 cu. ft. of free air per minute, to 650 ft. per minute for a large, 5-ft. stroke, two-stage compressor, driven by a compound Corliss engine with a rated capacity of 8000 cu. ft. of free air per minute.

56. The mechanical efficiency of a compressor is the ratio of the power theoretically required to compress and deliver a given quantity of air per unit of time to the power actually consumed.

The power in excess of the theoretical power is chiefly required to overcome frictional resistances of the machine, for which no return is made in the ultimate use of the compressed air. It therefore is a loss which can be minimized but not avoided altogether.

The amount of friction is dependent both on the design of the compressor and on the care bestowed upon it while in operation.

CHAPTER VI

TWO-STAGE AND MULTI-STAGE COMPRESSION, ALSO KNOWN AS COMPOUND COMPRESSION

THEORY

57. Single-stage, isothermal and adiabatic compression of air from 0 to 120 lb. gage is represented in Fig. 9 by the curves *AC* and *AE* respectively.

In preceding articles it was explained that the work performed during one stroke of the piston is represented by the area in the diagram to the right of the compression curve. It was also explained that the expenditure of energy in adiabatic compression and delivery is greater than in isothermal compression and delivery, on account of the increase in volume, due to the unchecked rise in temperature.

In the diagram, Fig. 9, the area *ADECBA* therefore stands for the waste of energy in adiabatic, single-stage compression and delivery, in which the heat of compression is allowed to remain in the air. This heat of compression represents work done upon the air for which there is no return, since during transmission the heat is all lost by radiation before the air is used.

The problem of economy, obviously, becomes one of abstracting the heat generated in the air during the process of compression. As has been pointed out, this is partially accomplished by water-jacketing of the cylinder walls. But owing to the short interval within which the compression takes place and the comparatively small volume of air actually in contact with the cylinder walls, very little cooling of the air occurs. Cylinder jackets are, however, indispensable in keeping the cylinder walls sufficiently cool for effective lubrication, and in the prevention of cumulative heating, which in extreme cases may result in explosion.

The impossibility of proper cooling within a single cylinder leads to the alternative of discharging the air from one cylinder, after partial compression has been effected, into a so-called

inter-cooler, removing the heat generated during the first compression, and then compressing the air to final pressure in another cylinder. This method of accomplishing compression in two steps with intermediate cooling is called two-stage compression, or when repeated one or more times for high pressures, multistage compression. See Figs. 24 to 33 for designs of modern two- and multi-stage compressors.

Referring again to diagram Fig. 9 and assuming the compression in each cylinder to be adiabatic, the compression curve is represented by the interrupted line $ADBH$; the compres-

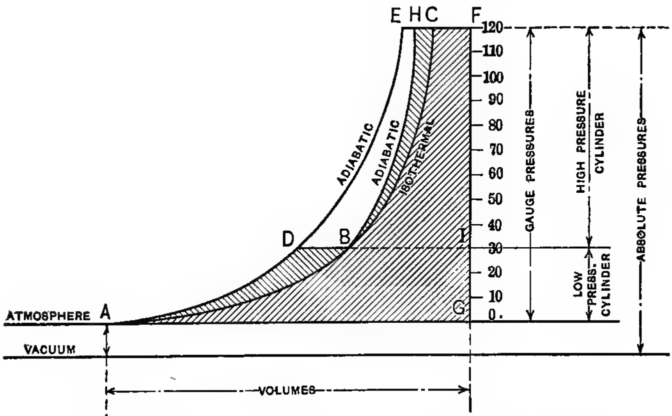


FIG. 9.—Diagram Illustrating Two-stage Compression.

sion proceeds adiabatically in the first or low-pressure cylinder to D; the air is then withdrawn and cooled under practically constant pressure in a suitable vessel called an inter-cooler, until its initial temperature is reached and its volume is reduced from ID to IB; with good inter-cooler arrangement it may be even further reduced; it is then introduced into a second or high-pressure cylinder and compressed adiabatically as before along the line BH to the final pressure.

As before noted, the energy required in single-stage compression and discharge of a given quantity of air under isothermal conditions is proportional to the shaded area $ABCFG$. The additional energy required in two-stage adiabatic compression is proportional to the other shaded areas ADB and BCH; while the loss of energy in adiabatic single-stage compression and delivery compared with isothermal compression and delivery is

proportional to the whole area $ADECBA$. The saving effected by two-stage adiabatic compression and delivery is therefore represented by the unshaded portion $DEHB$.

From Table V it appears that to compress 1 cu. ft. of free air per minute to 100 lb. gage and deliver it at that pressure into the receiver, requires 0.182 h.p. in single-stage and 0.158 h.p. in two-stage compression at sea level, showing a saving in energy of 13 per cent. in two-stage compression.

58. Analysis of Two-stage Compression.—Fig. 10 shows the low- and high-pressure air cylinders and the inter-cooler of

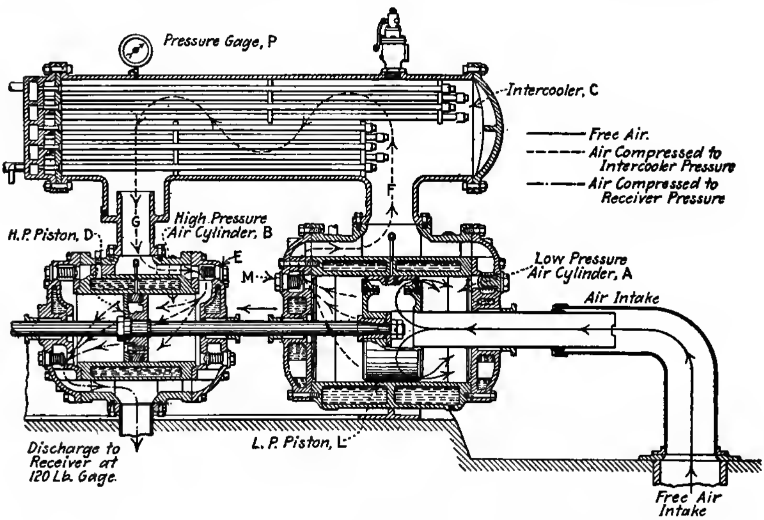


FIG. 10.—Diagram Illustrating Principle of Two-stage Compression.

an Ingersoll-Rand straight line steam-driven, two-stage air compressor.

In order to study the principle of operation of this compressor we must assume that it has been running long enough to bring about normal working conditions; viz., that the pressure in the receiver has reached a required terminal pressure of, say, 120 lb. gage and that the inter-cooler is therefore filled with air at about 30 lb. gage.

To follow the air through its various stages from the in-take to the discharge during one stroke of the piston, let it be assumed that the pistons advance from right to left as indicated by the arrow in Fig. 10. During the previous stroke from left to right

the inter-cooler (*C*) and its connections (*F* and *G*) to the air cylinders have been replenished with air compressed to 30 lb. This body of air is now shut off from both cylinders by their relative valves and is losing the greater part of its heat and some of its pressure through the influence of the circulating cold water in the inter-cooler. The loss of pressure is quickly made up by the equalizing process explained below.

During the first part of the return stroke from right to left the piston (*L*) in the low-pressure cylinder (*A*) acts only on the free air taken in on the previous stroke, while the high-pressure piston (*D*) is engaged in compressing to the terminal pressure the air in front of it, which has been admitted on the previous stroke from the inter-cooler at a pressure slightly under 30 lb.

While the free air in the low-pressure cylinder (*A*) is being compressed, the advance of the piston (*D*) in the high-pressure cylinder reduces the pressure behind it, causing the high-pressure inlet valves (*E*) to open. The compressed air in the inter-cooler (*C*) and in the connections (*F* and *G*) now rushes into the high-pressure cylinder (*B*) thereby slightly expanding in volume and decreasing in pressure until the pistons have reached a point somewhat beyond midstroke.

When the pistons have passed this point, the air pressure in front of the low-pressure piston (*L*) rises slightly higher than that in the inter-cooler, causing the low-pressure delivery valves (*M*) to open. From now on to the end of the stroke both cylinders are in communication with each other through the inter-cooler.

The low-pressure piston now acts upon the entire body of air contained in the low-pressure cylinder, in front of the piston, in the inter-cooler, in the connecting passages and in the portion of the high-pressure cylinder behind the high-pressure piston. At the same time the air in front of the high-pressure piston is delivered into the receiver at constant final pressure.

During this period an approximate equalization of pressure is established throughout. These fluctuations in pressure and the final equalization of pressure which take place during each stroke of the piston may be observed by watching the movements of the hand on the pressure gage (*P*) and from the indicator diagrams taken from compressors in normal operation. (See Chapter VIII.)

Referring to the diagram, Fig. 9, the saving of energy in two-stage compression was explained as being due to the inter-cooler which reduces the volume *ID* of the compressed air, coming

from the low-pressure cylinder, to a volume *IB*, before admitting it into the second or high-pressure cylinder.

In the above analysis of operation no mention is made of a reduction in volume. The reason for this becomes clear when it is realized that the air which enters the high-pressure cylinder from the inter-cooler is not the identical volume of air which leaves the low-pressure cylinder at that moment, but is a quantity of air that had been in the inter-cooler long enough to cool down to atmospheric temperature and, being cooled, occupies in the inter-cooler a smaller volume than it did when it entered it. The room made by the shrinkage is immediately filled at the other end of the inter-cooler by a quantity of hot air rushing into it from the low-pressure cylinder during the process of equalization mentioned above.

The above description of the mode of operation does not strictly apply to a duplex cross-compound compressor such as shown in Fig. 28 because there the pistons work with one crank 90 degrees in advance of the other, and at certain periods in the cycle of operation travel in opposite directions. The effect of the inter-cooler, however, is practically the same as in straight line machines.

RATIO OF COMPRESSION IN COMPOUND OR MULTI-STAGE COMPRESSION

59. The ratio of compression in any cylinder of a compressor is:

$$r = \frac{\text{terminal absolute pressure in that cylinder}}{\text{initial absolute pressure in that cylinder}}$$

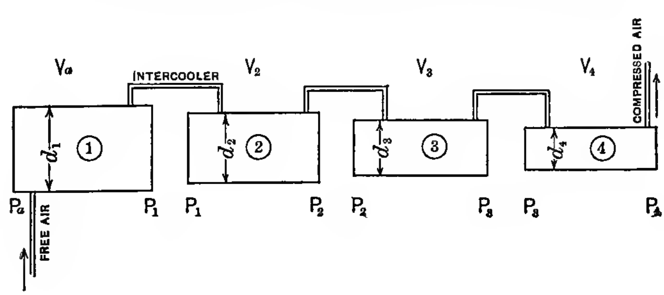


FIG. 11.

The cylinders of a multi-stage compressor are dimensioned so that the work done per stroke in each cylinder is as nearly as

possible the same, thereby equalizing and minimizing the strains on the machine.

Let 1, 2, 3, 4 in Fig. 11 represent the cylinders of a multi-stage compressor, and let

P_a = initial absolute pressure in cylinder (1) in pounds per square inch,
P_1 = final absolute pressure in cylinder (1) in pounds per square inch,
= initial absolute pressure in cylinder (2) in pounds per square inch,
P_2 = final absolute pressure in cylinder (2) in pounds per square inch,
= initial absolute pressure in cylinder (3) in pounds per square inch,

&c.

V_a = piston displacement in cylinder (1) in cubic feet,
V_2 = piston displacement in cylinder (2) in cubic feet,

&c.

then the net work per stroke of adiabatic compression and delivery in cylinder (1) is, according to Article 45:

$$W_n' = \frac{144n}{n-1} P_a V_a \left[\left(\frac{P_1}{P_a} \right)^{\frac{n-1}{n}} - 1 \right] \text{ foot-pounds.} \tag{1}$$

The net work per stroke in cylinder (2) is:

$$W_n'' = \frac{144n}{n-1} P_1 V_2 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{ foot-pounds.} \tag{2}$$

In compound compression the air between stages is supposed to be cooled to initial temperature under constant pressure; hence the volume V_2 of air, entering cylinder (2) from the inter-cooler at an absolute pressure P_1 is the volume which cylinder (1) would deliver per stroke, had it compressed the in-take air V_a isothermally during that stroke instead of adiabatically to the pressure P_1. From this it follows that in stage compression with perfect inter-cooling between stages, Boyle's law (Article 15) becomes applicable:

$$P_a V_a = P_1 V_2$$

Substituting this value in equation (2) we get

Net work per stroke in cylinder 2:

$$W_n'' = \frac{144n}{n-1} P_a V_a \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{ foot-pounds} \qquad (3)$$

Net work per stroke in cylinder 1:

$$W_n' = \frac{144n}{n-1} P_a V_a \left[\left(\frac{P_1}{P_a} \right)^{\frac{n-1}{n}} - 1 \right] \text{ foot-pounds.} \qquad (1)$$

A glance at equations (3) and (1) shows that W_n'' becomes equal to W_n' when the ratio of compression $\frac{P_2}{P_1}$ in cylinder (2) becomes equal to the ratio of compression $\frac{P_1}{P_a}$ in cylinder (1).

This leads to the general conclusion: In order that the work per stroke in each cylinder of a compound compressor be theoretically the same, the following relations between the terminal and initial pressures in the various cylinders must exist:

$$\frac{P_1}{P_a} = \frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} \text{ etc.}$$

60. For two-stage compression we have:

$$\frac{P_1}{P_a} = \frac{P_2}{P_1} \qquad (1)$$

Dividing by P_a

$$\frac{P_1^2}{P_a^2} = \frac{P_2}{P_a}$$

Whence

$$\frac{P_1}{P_a} = \frac{P_2}{P_1} = \sqrt{\frac{P_2}{P_a}} \qquad (2)$$

This shows that the ratio of compression in each of the two cylinders must be equal to the square root of the total ratio of compression; the total ratio being the ratio between the final pressure in cylinder (2) and the initial pressure in cylinder (1).

From equation (2) we get:

$$P_1 = P_a \sqrt{\frac{P_2}{P_a}} = \sqrt{P_a P_2} \qquad (3)$$

4

61. For three-stage compression we have:

$$\frac{P_1}{P_a}=\frac{P_2}{P_1}=\frac{P_3}{P_2} \tag{1}$$

Whence
$$P_2=\frac{P_1^2}{P_a}$$

Substituting this value in (1)

$$\frac{P_1}{P_a}=\frac{P_3P_a}{P_1^2}$$

Dividing by P_a^2,

$$\frac{P_1^3}{P_a^3}=\frac{P_3}{P_a}$$

Whence
$$\frac{P_1}{P_a}=\frac{P_2}{P_1}=\frac{P_3}{P_2}=\sqrt[3]{\frac{P_3}{P_a}} \tag{2}$$

That is, the ratio of compression in each of the three cylinders must be equal to the cube root of the total ratio of compression.

From equation (2) we get

$$P_1=P_a\sqrt[3]{\frac{P_3}{P_a}}=\sqrt[3]{P_a^2P_3} \tag{3}$$

$$P_2=P_1\sqrt[3]{\frac{P_3}{P_a}}=\sqrt[3]{P_a^2P_3}\times\sqrt[3]{\frac{P_3}{P_a}}=\sqrt[3]{P_aP_3^2} \tag{4}$$

62. For four-stage compression we would find

$$\frac{P_1}{P_a}=\frac{P_2}{P_1}=\frac{P_3}{P_2}=\frac{P_4}{P_3}=\sqrt[4]{\frac{P_4}{P_a}} \tag{1}$$

whence
$$P_1=P_a\sqrt[4]{\frac{P_4}{P_a}}=\sqrt[4]{P_a^3P_4} \tag{2}$$

$$P_2=P_1\sqrt[4]{\frac{P_4}{P_a}}=\sqrt[4]{P_a^3P_4}\times\sqrt[4]{\frac{P_4}{P_a}}=\sqrt[4]{P_a^2P_4^2}=\sqrt{P_aP_4} \tag{3}$$

$$P_3=P_2\sqrt[4]{\frac{P_4}{P_a}}=\sqrt[4]{P_a^2P_4^2}\times\sqrt[4]{\frac{P_4}{P_a}}=\sqrt[4]{P_aP_4^3} \tag{4}$$

CYLINDER DIAMETERS OF MULTI-STAGE COMPRESSORS

63. The cylinders of a multi-stage compressor are proportioned in accordance with the initial volume V_a of free air to be com-

pressed per stroke or per minute and the ratio of compression in each cylinder, as obtained from the formulas in Articles 60, 61 and 62.

In compressed-air computations the volume V_a to be compressed per stroke or per minute is usually given; the length of the stroke is made the same in all cylinders; the number of strokes per minute is chosen with reference to the type of compressor desired and the number of cubic feet of air required per minute.

Referring to Fig. 11:

Let V_a = volume of free air in cubic feet, taken into cylinder (1) per stroke,

V_2 = volume which the air discharged from cylinder (1) occupies at a pressure P_1 after being cooled to initial temperature,

= piston displacement of cylinder (2),

V_3 = volume which the air discharged from cylinder (2) occupies at a pressure P_2 after being cooled to initial temperature,

= piston displacement of cylinder (3),

&c.

d_1 = diameter of cylinder (1) in inches,

d_2 = diameter of cylinder (2) in inches,

&c.

A = area of piston in cylinder (1) in square inches,

L = length of stroke in inches,

then
$$V_a = \frac{A}{144} \times \frac{L}{12} = \frac{0.7854 d_1^2 L}{1728}$$

whence
$$d_1 = 47\sqrt{\frac{V_a}{L}} \text{ inches.}$$

Having thus determined the diameter d_1 of the in-take or low-pressure cylinder (1), the diameters of the other cylinders are found as follows:

The length of stroke being the same for each cylinder, the volumes of the cylinders are in the ratio of the squares of their diameters. Assuming complete inter-cooling between stages, the volumes according to Article 16, are also in the inverse ratio of the pressures. In this connection it must be remembered

that volume V_2, for instance, is the volume which the air discharged from cylinder (1) would occupy after being cooled to initial temperature under constant pressure P_1.

Referring to Fig. 11 we would have

$$\frac{d_2^2}{d_1^2}=\frac{V_2}{V_a}=\frac{P_a}{P_1}=\frac{1}{\frac{P_1}{P_a}}$$

hence

$$\frac{\text{square of diam. of one cylinder}}{\text{square of diam. of preceding cyl.}}=\frac{1}{\text{ratio of compr. in each cyl.}}=\frac{1}{r}$$

From Articles 60 to 62 we have:

For two-stage compression:

$$\frac{1}{r}=\frac{1}{\left(\frac{P_2}{P_a}\right)^{\frac{1}{2}}}=\left(\frac{P_a}{P_2}\right)^{\frac{1}{2}}$$

For three-stage compression:

$$\frac{1}{r}=\frac{1}{\left(\frac{P_3}{P_a}\right)^{\frac{1}{3}}}=\left(\frac{P_a}{P_3}\right)^{\frac{1}{3}}$$

For four-stage compression:

$$\frac{1}{r}=\frac{1}{\left(\frac{P_4}{P_a}\right)^{\frac{1}{4}}}=\left(\frac{P_a}{P_4}\right)^{\frac{1}{4}}$$

Therefore:

Cylinder Diameters

64. Cylinder Diameters for a Two-stage Compressor.—

$$d_1=47\sqrt{\frac{V_a}{L}}\text{ inches}$$

$$\frac{d_2^2}{d_1^2}=\left(\frac{P_a}{P_2}\right)^{\frac{1}{2}}\qquad\text{whence}\qquad d_2=d_1\left(\frac{P_a}{P_2}\right)^{\frac{1}{4}}\text{ inches}$$

65. Cylinder Diameters for a Three-stage Compressor.—

$$d_1 = 47\sqrt{\frac{V_a}{L}} \text{ inches}$$

$$\frac{d_2^2}{d_1^2} = \left(\frac{P_a}{P_3}\right)^{\frac{1}{3}} \quad \text{whence} \quad d_2 = d_1\left(\frac{P_a}{P_3}\right)^{\frac{1}{6}} \text{ inches}$$

$$\frac{d_3^2}{d_2^2} = \left(\frac{P_a}{P_3}\right)^{\frac{1}{3}} \text{ or } d_3 = d_2\left(\frac{P_a}{P_3}\right)^{\frac{1}{6}}$$

$$d_3 = d_1\left(\frac{P_a}{P_3}\right)^{\frac{1}{6}} \times \left(\frac{P_a}{P_3}\right)^{\frac{1}{6}} \quad \text{whence} \quad d_3 = d_1\left(\frac{P_a}{P_3}\right)^{\frac{1}{3}} \text{ inches}$$

66. Cylinder Diameters for a Four-stage Compressor.—

$$d_1 = 47\sqrt{\frac{V_a}{L}} \text{ inches}$$

$$\frac{d_2^2}{d_1^2} = \left(\frac{P_a}{P_4}\right)^{\frac{1}{4}} \quad \text{whence} \quad d_2 = d_1\left(\frac{P_a}{P_4}\right)^{\frac{1}{8}} \text{ inches}$$

$$\frac{d_3^2}{d_2^2} = \left(\frac{P_a}{P_4}\right)^{\frac{1}{4}} \text{ or } d_3 = d_2\left(\frac{P_a}{P_4}\right)^{\frac{1}{8}}$$

$$d_3 = d_1\left(\frac{P_a}{P_4}\right)^{\frac{1}{8}} \times \left(\frac{P_a}{P_4}\right)^{\frac{1}{8}} \quad \text{whence} \quad d_3 = d_1\left(\frac{P_a}{P_4}\right)^{\frac{1}{4}} \text{ inches}$$

$$\frac{d_4^2}{d_3^2} = \left(\frac{P_a}{P_4}\right)^{\frac{1}{4}} \text{ or } d_4 = d_3\left(\frac{P_a}{P_4}\right)^{\frac{1}{8}}$$

$$d_4 = d_1\left(\frac{P_a}{P_4}\right)^{\frac{1}{4}} \times \left(\frac{P_a}{P_4}\right)^{\frac{1}{8}} \quad \text{whence} \quad d_4 = d_1\left(\frac{P_a}{P_4}\right)^{\frac{3}{8}} \text{ inches}$$

67. The volumetric efficiency of the compressor only affects cylinder (1). Therefore in the above calculations of the diameters d_1, d_2, d_3, etc., the value d_1 as given in the formula will be used, and the final value of d_1 will be d_1 as found by the formula, plus an allowance, depending on the volumetric efficiency of the compressor.

Example *a.*—What should be the diameters of the air-cylinders of a three-stage compressor to furnish 300 cu. ft. of free air per minute, compressed to 300 lb. gage?

Length of stroke $L = 16$ in.
Number of r.p.m. $= 135$
Volumetric efficiency $= 85$ per cent.
Diameter of piston rod $= 2\ 1/2$ in.

b. What is the terminal pressure in each of the cylinders and what is the ratio of compression in each cylinder?

Solution *a*. The number of revolutions being 135, the number of strokes per minute is 270. Hence the volume of free air taken into the cylinder per stroke is:

$$\frac{300}{270} = 1.11 \text{ cu. ft.}$$

To which must be added, volume of piston rod·

$$\frac{16 \times 2.5^2 \times 0.7854}{1728} = 0.045 \text{ cu. ft.}$$

$$\text{Hence } V_a = 1.11 + 0.045 = 1.155 \text{ cu. ft.}$$

$$\text{And} \quad d_1 = 47\sqrt{\frac{1.155}{16}} = 12.63 \text{ in.}$$

$$d_2 = d_1\left(\frac{14.7}{300+14.7}\right)^{\frac{1}{6}} = 7.58 \text{ in.}$$

$$d_3 = d_1\left(\frac{14.7}{300+14.7}\right)^{\frac{1}{3}} = 4.55 \text{ in.}$$

The diameter d_1 of the in-take cylinder must be increased to allow for a volumetric efficiency of 85 per cent.

Calling the final diameter of the in-take cylinder x we have:

$$\frac{x^2}{d_1^2} = \frac{100}{85}$$

$$\text{Whence} \quad x = d_1\sqrt{\frac{100}{85}} = 12.63 \times 1.085 = 13.71 \text{ in.}$$

Solution *b*. From Article 61, terminal pressure in

Cylinder (1) $P_1 = \sqrt[3]{14.7^2 \times 314.7} =$ 40.81 lb. absolute.
$=$ 26.11 lb. gage.

Cylinder (2) $P_2 = \sqrt[3]{14.7 \times 314.7^2} = 113.34$ lb. absolute.
$=$ 98.64 lb. gage.

Cylinder (3) P_3 $= 314.7$ lb. absolute
$= 300.00$ lb. gage.

Ratio of compression in each cylinder:

$$\sqrt[3]{\frac{P_3}{P_a}}=\sqrt[3]{\frac{300+14.7}{14.7}}=2.78$$

In cylinder (1) $\qquad \dfrac{40.81}{14.7}=2.78$

In cylinder (2) $\qquad \dfrac{113.34}{40.81}=2.78$

In cylinder (3) $\qquad \dfrac{314.7}{113.34}=2.78$

THEORETICAL HORSE-POWER, COMPOUND COMPRESSION AND DELIVERY

68. For two-stage compression, the theoretical horse-power required to compress adiabatically a volume of free air in cubic feet per minute from an initial absolute pressure P_a to a final pressure P_2 and deliver it at that pressure into the receiver, is the sum of the horse-power required in each of the two cylinders. In Article 60 it was shown that for a two-stage compressor the ratio of compression in each cylinder must be equal to the square root of the total ratio of compression, that is:

$$r=\left(\frac{P_2}{P_a}\right)^{\frac{1}{2}}$$

The horse-power required in each of the two cylinders is therefore

$$\text{Horse-power}=\frac{144\,nV_aP_a}{33{,}000\,(n-1)}\left[\left(\frac{P_2}{P_a}\right)^{\frac{1}{2}\times\frac{n-1}{n}}-1\right]$$

And for the two cylinders the horse-power is just twice that amount.

69. Theoretical horse-power—two-stage compression and delivery.

$$\text{Horse-power}=2\frac{144\,n\,V_aP_a}{33{,}000\,(n-1)}\left[\left(\frac{P_2}{P_a}\right)^{\frac{n-1}{2n}}-1\right]$$

in which V_a = volume of free air in cubic feet per minute, taken into the low-pressure cylinder (1).

P_a = initial absolute pressure in pounds per square inch in the low-pressure cylinder (1).

P_2 = final absolute pressure in pounds per square inch in the high-pressure cylinder (2).

n = exponent of adiabatic compression ($=1.406$).

In the same manner we find for:

70. Three-stage compression

$$\text{Horse-power} = 3\frac{144\ nV_aP_a}{33{,}000\ (n-1)}\left[\left(\frac{P_3}{P_a}\right)^{\frac{n-1}{3n}} - 1\right]$$

71. Four-stage compression

$$\text{Horse-power} = 4\frac{144\ nV_aP_a}{33{,}000\ (n-1)}\left[\left(\frac{P_4}{P_a}\right)^{\frac{n-1}{4n}} - 1\right]$$

The theoretical mean gage pressure in pounds per square inch is for:

72. Two-stage compression

$$P_m = 2\frac{n}{n-1}P_a\left[\left(\frac{P_2}{P_a}\right)^{\frac{n-1}{2n}} - 1\right]$$

73. Three-stage compression

$$P_m = 3\frac{n}{n-1}P_a\left[\left(\frac{P_3}{P_a}\right)^{\frac{n-1}{3n}} - 1\right]$$

Columns 7, 10, and 13 of Table V give the theoretical horse-power required at sea level to compress adiabatically and deliver 1 cu. ft. of free air per minute by two-, three-, and four-stage compression.

FINAL VOLUMES AND TEMPERATURES OF AIR IN STAGE COMPRESSION

73A. Two-stage Compression.—On its passage from the low-pressure cylinder via the intercooler to the high-pressure cylinder, the air is cooled under constant pressure to initial temperature. On entering the high-pressure cylinder its volume V_1 will therefore be equal to a volume, the intake-air V_a would occupy had it been compressed isothermally in the low-pressure cylinder. According to Boyle's law this volume is

$$V_1 = V_a\frac{P_a}{P_1} \tag{1}$$

In the high-pressure cylinder volume V_1 is compressed adiabatically to a volume V_2 and to an absolute pressure P_2.

Therefore $$\frac{V_2}{V_1}=\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

Introducing value of V_1 from (1) and value

$$P_1=\sqrt{P_aP_2} \text{ from (3), Art. 60,}$$

we get $$V_2=V_a\left(\frac{P_a}{P_2}\right)^{\frac{n+1}{2n}} \qquad (2)$$

The absolute temperature of the discharge air will be the same as temperature T_1 of the air when leaving the low-pressure cylinder, the ratio of compression being the same in both cylinders.

According to (11), Art. 41

$$T_2=T_a\left(\frac{P_1}{P_a}\right)^{\frac{n-1}{n}}=T_1$$

Introducing $$P_1=\sqrt{P_aP_2}$$

we get $$T_2=T_a\left(\frac{P_2}{P_a}\right)^{\frac{n-1}{2n}} \qquad (3)$$

By similar deductions we obtain analogous values for three- and four-stage compression as indicated in Table VIII.

74. Modified Power Values for Practical Problems.—In the preceding theoretical formulas no allowance has been made for clearance, the heating of the in-take air and the friction of the compressor. As previously stated, the first two items are negligible as far as they affect power consumption. For friction an additional allowance of from 7 to 15 per cent. of the theoretical horse-power is usually made in practical compressor calculations.

75. Advantages of Multi-stage Compression.—The principal advantage of compound or multi-stage compression over single-stage compression lies in the saving of energy by reducing the heat of compression as pointed out above. Other important advantages due to compounding may be summed up as follows:

a. Reduced Strain on Machine.—This will appear from the following illustration:

Consider two compressors, compressing air to 120 lb. gage, one compressor having a single air cylinder of the usual pattern,

the other having compound cylinders. With a piston of 100 sq. in. in area, the maximum resistance which the single-stage compressor must overcome, would be 12,000 lb.

Let us now consider a two-stage compressor in which the area of the low-pressure cylinder piston is again 100 sq. in. and that of the high-pressure cylinder one-third, or 33 1/3 sq. in. In the low-pressure cylinder the air is compressed to about 30 lb. Since this pressure of 30 lb. acts on the back of the piston in the high-pressure cylinder, it assists the machine, and the net resistance of forcing the air from the larger into the smaller cylinder is equal to the difference in the areas of the two pistons (which is 66 2/3 sq. in.) multiplied by 30 lb. This equals 2000 lb.

In the smaller or high-pressure cylinder the maximum resistance to overcome is $100 \times 33\ 1/3 = 3333$ lb., and the sum of the two resistances at the time of greatest effort in the two-stage compressor is 5333 lb. as compared with 12,000 lb. in the single-stage compressor, representing a reduction in strains of more than one-half.

b. Improved Steam Economy.—The more equable distribution of the load throughout the stroke greatly reduces the danger of centering. This permits an earlier cut-off in the steam cylinder, resulting in a greater steam expansion. With properly designed inter-coolers the piston speed can be increased without danger of overheating the cylinders. Increased piston speed is in itself a factor in steam economy, since it reduces leakage and condensation in the steam end of the compressor.

c. Increased Safety and Ease of Lubrication.—When high final temperatures prevail, part of the lubricating oil vaporizes, and the wear on piston and cylinder becomes rapid. Under exceptional circumstances the combination of air and oil vapor and other combustibles may reach the proportions of an explosive mixture and, if the compression temperature reaches its ignition point, an explosion may result. Such accidents are, however, very rare even in single-stage work; in multi-stage compression, with proper inter-cooling and proper attention they are practically impossible.

If the work of compression has been divided equally between the cylinders by a correct proportioning of their diameters, and if the inter-coolers are properly designed, the final temperature in each cylinder will be the same, and it will be much lower than if compression was completed in one cylinder. *To illustrate:*

In compressing air at atmospheric temperature of 60° Fahr. to 100 lb. pressure in a two-stage compressor, the air is compressed from atmospheric pressure to 26 1/2 lb. in the in-take or low-pressure cylinder, and is delivered to the inter-cooler at this pressure and at 240° Fahr. If all the heat of compression is taken out by the inter-cooler, it is admitted to the high-pressure cylinder at atmospheric temperature and is then compressed from 26 1/2 lb. to 100 lb. and delivered to the receiver at a temperature of 240° Fahr. (Radiation and cooling by water-jackets not considered.)

In a single-stage compressor the air is compressed from atmospheric pressure to 100 lb. in one cylinder and reaches the receiver at a temperature of 482° Fahr.

d. Greater Effective Capacity in Free Air.—Clearance loss in an air compressor is principally a loss in capacity, and affects only the in-take cylinder; it increases with the terminal pressure in this cylinder. Since in compound compression the terminal pressure in the low-pressure cylinder is much lower than in the single-stage machine, the air confined in the clearance spaces, when expanded down to atmospheric pressure, occupies comparatively little space. Consequently the in-flow of air through the suction or inlet valves begins at an earlier point in the stroke than it would in the single-stage compressor, which results in a greater volumetric efficiency of the compound compressor. (See discussion of indicator cards in Chapter VIII.)

e. Dryer Air.—The air delivered by a compound compressor is dryer than that furnished by a single cylinder. Under constant pressure, the power of air to hold vapor decreases with its temperature, and during its passage through the inter-cooler much of the original moisture in the air is precipitated. Consequently less trouble is experienced from condensation in the discharge pipe, and the danger of freezing up the exhaust ports of machines using compressed air is greatly reduced.

76. When to use Two- and Multi-stage Compression.—Below and up to 60 or 70 lb. terminal pressure, the adiabetic loss is comparatively trival, and within this limit and at low altitudes, single-stage compressors are commonly employed. Between 60 and 100 lb., the amount of fuel is usually the determining factor, though high altitude may also enter into the question. Above 100 lb. both safety and economy speak for two-stage and above 500 lb., for multi-stage compressors.

CHAPTER VII

EFFECT OF ALTITUDE ON AIR COMPRESSION

77. Volumetric Efficiency.—The volumetric efficiency of a compressor, expressed *in terms of free air*, is the same at all altitudes because the piston displacement in a cylinder of a given size is the same. But the volumetric efficiency, expressed *in terms of compressed air*, decreases as the altitude increases.

Since the density and hence the atmospheric pressure decreases with the altitude, a compressor located at an altitude above sea level takes in at each revolution a smaller weight of air at a lower pressure than at sea level, and the early part of each stroke is occupied in compressing the air from this lower pressure up to the sea level pressure. In other words, the free air taken into a cylinder per stroke being less dense at an altitude (due to lower initial atmospheric pressure) it will be compressed into a smaller space for a given terminal pressure.

Example.—Five-hundred cubic feet of air at atmospheric pressure at sea level (14.7 lb.), compressed isothermally to 80 lb. gage, occupies a volume of

$$500\times\frac{14.7}{80+14.7}=77.6 \text{ cu. ft.}$$

From Table VI the atmospheric pressure at an altitude of 10,000 ft. is 10.07 lb. and 500 cu. ft. of air, compressed isothermally to 80 lb. gage at that altitude would occupy a volume of

$$500\times\frac{10.07}{80+10.07}=55.9 \text{ cu. ft.}$$

That is, the volumetric efficiency *in terms of compressed air* of a compressor performing the above work at an altitude of 10,000 ft. is only 72 per cent. of what it would be at sea level.

In order, therefore, that an air compressor at an altitude may deliver a volume of compressed air per stroke equal to that which it would deliver at sea level, the in-take cylinder of the altitude compressor must be proportionally larger than that of a compressor at sea level.

78. Multipliers for Altitude Computations.—Referring to the preceding example, multipliers may be computed for determin-

ing the volume of free air at various altitudes which, when compressed to various pressures, is equivalent in effect to a given volume of free air compressed to the same pressure at sea level.

Let V = a certain number of cubic feet of atmospheric air to be compressed simultaneously at sea level and at an elevation above sea level,

P_a = absolute pressure of atmospheric air in lbs. per square inch at sea level (14.75 lb.),

P_1 = absolute pressure of atmospheric air in lbs. per square inch at the given elevation,

p = gage pressure to which air is being compressed,

then the volume V_1 which the air occupies after being compressed to p pounds gage at sea level:

$$V_1 = V \frac{P_a}{(p+P_a)}$$

And the volume V_2 which the air occupies after being compressed to p pounds gage at the elevation:

$$V_2 = V \frac{P_1}{(p+P_1)}$$

In order that V_2 may be equal to V_1 it must be multiplied by a multiplier "M" which we find as follows:

$$M V_2 = V_1$$

Substituting values

$$M V \frac{P_1}{p+P_1} = V \frac{P_a}{p+P_a}$$

Whence
$$M = \frac{P_a(p+P_1)}{P_1(p+P_a)}$$

Example.—What is the multiplier for a volume of air at 5000 ft. elevation and for a pressure of 80 lb. gage. (See Table VI.)

$$M = \frac{P_a(p+P_1)}{P_1(p+P_a)} = \frac{14.75\ (80+12.20)}{12.20\ (80+14.75)} = 1.178$$

If for instance we wish to know the volume of free air which after being compressed to 80 lb. gage at an altitude of 5000 ft., has the same effect as, say 100 cu. ft. of air compressed to 80 lb. gage at sea level, we find it by multiplying 100 by 1.178.

Thus, $100 \times 1.178 = 117.8$ cu. ft.

79. Power Required for Altitude Compressors.—To compress a given volume of free air taken in by a compressor of given size to a given terminal pressure takes *less power* at an altitude than at sea level. The air being lighter and less dense, its volume at the desired terminal pressure will be smaller, that is, the final pressure is reached at a later point in the stroke. Hence the mean pressure is less and so is the total power required to compress the quantity of air taken into the cylinder.

But, in order to produce at an altitude a quantity of compressed air which is equivalent in effect to air at sea level, *more power* is required, because the reduction in power referred to above is not proportional to the increase in volume necessary to equal sea-level performance.

Example.—Using formula, Article 68, we find that to compress by two-stage compression 100 cu. ft. of free air per minute at sea-level pressure (14.7 lb.) to 100 lb. gage and deliver it into the receiver requires 15.80 h.p. At an altitude of 10,000 ft., where the atmospheric pressure is 10.07 lb., it would require only 12.50 h.p.

But from Table VI, a quantity of air which at an elevation of 10,000 ft. is the equivalent of 100 cu. ft. of free air at sea level would occupy a volume of 140.4 cu. ft. To compress this quantity to 100 lb. gage and deliver it into the receiver would require:

$$\frac{140.4}{100}\ 12.50 = 17.55 \text{ h.p.}$$

This shows that in this case 1.75 additional horse-power are required at 10,000 ft. elevation to produce the same effect as at sea level.

80. Stage Compression at High Altitudes.—From what has been said of the effect of altitude on air compression, it becomes evident that stage compression at altitudes results in even larger percentage of saving of power than is possible at sea level.

Referring to Table V, it requires 0.182 h.p. at sea level to compress in one stage 1 cu. ft. of free air per minute to 100 lb. gage and deliver it into the receiver. Two-stage compression would consume only 0.158 h.p. which means a saving of 0.024 h.p. or of 13 per cent. in favor of two-stage compression.

At 9000 ft. above sea level the equivalent of 1 cu. ft. of air at sea level is 1.356 cu. ft. and the atmospheric pressure is 10.46 lb. (from Table VI). Horse-power required to compress 1.356 cu. ft. of free air per minute to 100 lb. gage at 9000 ft. elevation and deliver it into the receiver:

For single-stage compression, 0.21 h.p. (from Article 47)
For two-stage compression, 0.17 h.p. (from Article 68)

which means a saving of 0.04 h.p. or of 19 per cent. in favor of two-stage compression at an altitude of 9000 ft.

81. It has been pointed out heretofore that the volumetric efficiency of a compressor is higher in two-stage than in single-stage compression, owing to the smaller volume which the expanded clearance air occupies. This volume being a function of the ratio of compression, it follows that at high altitudes, stage compression can be profitably used for lower terminal pressures than is customary at sea level.

To illustrate: The ration of compression at sea level in compressing to 90 lb. gage is

$$\frac{90+14.7}{14.7}=7.12$$

The same ratio would obtain at an elevation of 10,000 ft. in compressing to 61.63 lb. gage; for

$$\frac{61.63+10.07}{10.07}=7.12,$$

which means that, if it pays to use two-stage compression at sea level in order to reduce clearance losses when compressing to 90 lb. gage, it would pay to do so at 10,000 ft. elevation, when compressing to 62 lb. gage.

The theoretical power required to compress a certain quantity of air at an altitude above sea level can be deduced from the formulas for the mean gage pressure and for the horse-power given under Adiabatic Compression (Articles 46, 47 and 68 to 72) by substituting for P_1 or P_a, respectively, the atmospheric pressure at the given altitude. The latter may be found from Table VI or calculated from formula (2), Article 4.

Manufacturers usually build special compressors for high altitudes which are designed to meet the demands made on a plant, located at considerable elevation above sea level.

CHAPTER VIII

THE COMPRESSED AIR INDICATOR CARD

82. Inasmuch as the work performed in the air cylinder of a compressor depends on so many variable and interdependent conditions, it can only be studied successfully from an indicator card, or still better from a number of indicator cards taken from the air cylinders of a compressor when in actual service.

It is assumed that the reader is familiar with the methods of taking an indicator card and of calculating from it the mean pressure, the horse-power, and the volumetric efficiency. There are a number of excellent books available which discuss this subject in detail, to which the reader is referred.

Besides the facts regarding power consumption, indicator diagrams also convey information regarding the working of valves, the volume of air taken in and compressed, the effect of clearance, the efficiency of the cooling devices, and the correct or incorrect proportioning of the air cylinders.

Unfortunately, indicator cards do not register temperatures and thus offer no means for determining directly the useful capacity of the compressor, expressed in number of pounds of free air at outside temperature compressed and delivered per unit of time. The latter is the only correct estimate of compressor capacity for the reason that before the air is used in the air engine, it has assumed outside temperature and any increase in volume, due to heat which the air may have received before or during compression, is only temporary and is subsequently lost for useful purposes.

Therfore an indicator card which shows high volumetric efficiency is in itself no proof of the ultimate merits of a compressor as far as capacity is concerned.

In Article 53 it was shown that inlet valves, which prevent ready admission of air, reduce the volumetric efficiency. This is shown on the air card by the suction line falling considerably below the atmospheric line. On the other hand, an air card taken from a compressor with leaky inlet valves may show a

large volumetric efficiency due to the fact that some of the clearance air escapes through them into the atmosphere and the expansion line *CG* (Fig. 8) falls sooner, thus increasing the distance *RG* without, however, increasing the useful quantity of air compressed.

In the same manner will leaky discharge valves, leaky pistons and inter-coolers show an apparent increase in volumetric efficiency while the actual quantity of air delivered is diminished.

Air cards, showing practically isothermal compression lines, are frequently the result of a leaky piston and not of superior workmanship in the construction of the compressor. A leaky piston will reduce the compression line as fully or more effectually than any cooling of the air can do, so that anyone not familiar with the tricks of an air card may easily be misled in his judgment of the merits of a machine from the indicator card alone.

Evidence of a leaky piston is usually given by a card showing the admission line above the atmospheric line during the entire stroke.

From what has been said, it is evident that great caution must be exercised in interpreting an air card. To do this correctly, it is necessary to know what positive information an air card will give and to bear in mind that it does not tell the whole story.

In the following articles a few air cards and their interpretation are given with the object of assisting the student of this subject in reading and interpreting similar cards.

83. Air Card of a Single-stage Compressor.—Fig. 12 is the facsimile of an indicator card taken from the air cylinder of a single-stage compressor. On this card the actual compression curve *AC* lies between the adiabatic curve *AB* and the isothermal curve *AD*. The gage pressure of the air in pounds per square inch at any point *M* of the stroke is measured by the line *MN*. The card shows the usual performance of a compressor of this type. When the piston has reached point *C* the air has reached receiver pressure (61 lb. gage on the card illustrated).

Owing to the weight of the discharge valves and the tension of the springs, the pressure in the cylinder usually rises a few pounds above the receiver pressure before the valves open, as shown at *E*. From this point to the end of the stroke the pressure drops to the receiver pressure. The wavy shape of the delivery line is mainly due to the fluttering of the discharge valves.

At the end of the forward stroke at *F* the piston comes to a

standstill. The discharge valves close and as soon as the piston commences the return stroke, the compressed air that was left in the clearance space begins to expand until it reaches atmospheric pressure at *G*. At this moment the inlet valves should open, but they are usually held for a moment against their seats by the tension of the springs. A partial vacuum is thus created behind the receding piston, causing the expansion line *FG* to drop below the atmospheric line as shown at *O*, whence it returns to the atmospheric line as soon as the valves open.

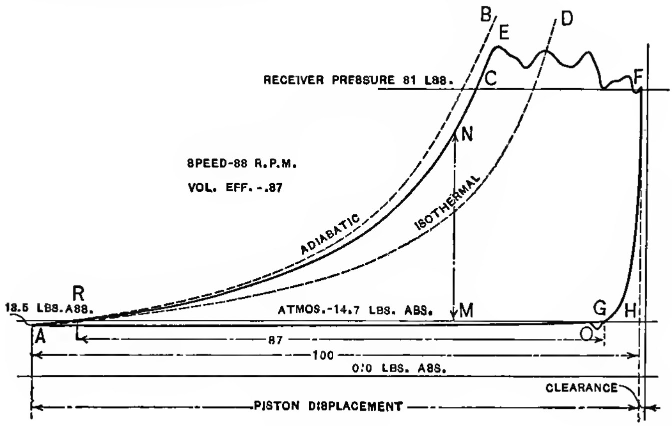


FIG. 12.

Free air is now admitted and if the inlet area is not restricted, the admission line will closely follow the atmospheric line from *G* to *A*. A considerable drop below the atmospheric line indicates that air is not admitted as freely as should be. This may happen when the slowing down of the piston at the end of the stroke permits the springs to close the inlet valves before the piston has completed its stroke. This reduces the volumetric efficiency of the compressor to the volume *GR*, because on the next forward stroke the piston must travel a distance *AR* before the atmospheric line is reached and before actual compression begins.

The line *HG* measures the extra volume taken up by the clearance air after expansion. It represents a loss which can be minimized by reducing the clearance space but cannot be avoided altogether. Theoretically, the loss is one of capacity only and not of power; for, although this air required work in compressing

it to receiver pressure, in expanding it helps to compress the air on the other side of the piston. (See Article 52.)

84. Air Card of a Two-stage Compressor.—Fig. 13 shows the combined cards taken from the crank-end of a two-stage compressor.

Low-pressure air cylinder	32 1/2×48 in.
High-pressure air cylinder	20 1/4×48 in.
Piston speed	480 ft. per minute.
Piston displacement per stroke	22.13 cu. ft.
Pressure of inlet air	14.00 lb. absolute.
Discharge pressure	78.00 lb. gage.
Volumetric efficiency (from card)	95 per cent.
Actual free air (from card)	2519 cu. ft. per minute.
Air horse-power (from card)	416.

The theoretical horse-power required for single-stage compression of the same quantity of air is 384, showing an excess of power consumed over a single-stage compression, amounting to 8.5 per cent.

This card, which is one of many that may be taken from otherwise well-built two-stage compressors, shows that in actual practice two-stage compression does not always result in a saving of power as would appear from theoretical calculations.

Whenever there is an overlap of the indicator cards taken from the low- and high-pressure cylinders of an air compressor, or when they run above the discharge pressure or materially below the in-take (atmospheric) pressure, there is power used in excess of the saving due to inter-cooling. Two-stage compressors giving cards such as shown in Fig. 13 fail to realize a saving of power over single-stage compressors usually through one or the other of the following defects.

a. Cylinders not properly proportioned for the prevailing compression ratio. In the case illustrated, the high-pressure cylinder seems to be too small and the low-pressure cylinder has to perform too much of the compression work.

b. Inter-cooler too small or inefficient. The theoretical saving of power is based on perfect cooling of the air to initial temperature after leaving the low-pressure cylinder. Failure on the part of the inter-cooler to do this will increase the work in the high-pressure cylinder.

c. Valve areas too small and air passages restricted. Re-

stricted inlet valve area in low-pressure cylinder not only increases the work to be done but also reduces the compressor efficiency. Restrictions in the high-pressure inlet, and low-pressure discharge valve areas, and in the inter-cooler and its piping, produce an excess in the discharge pressure from low-pressure cylinder to inter-cooler and a depression in the suction line of high-pressure indicator cards. The result of this is a

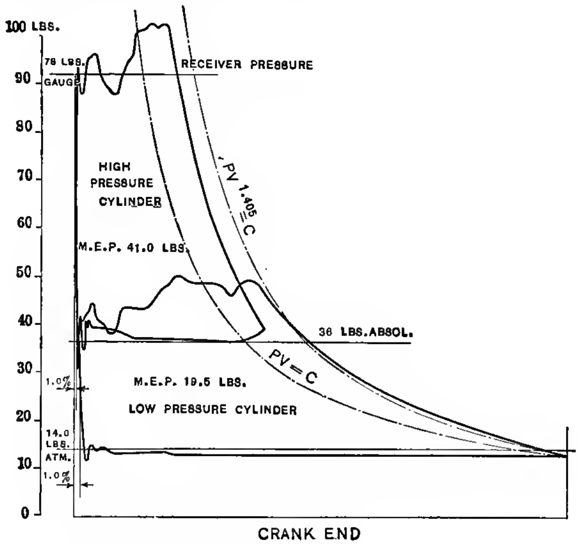


FIG. 13.—Combined Air Cards of a Two-stage Compressor.

great overlap in the developed diagrams where the high- and low-pressure cards come together. These pressure losses also produce an unequal distribution of the work in the two cylinders.

From Article 60 the terminal pressure in the low-pressure cylinder should have been:

$$P_1 = \sqrt{P_a P_2} = \sqrt{14.0(78+14)} = 36 \text{ lb. absolute.}$$

A glance at the diagram shows that the discharge valves of the low-pressure cylinder did not open until the pressure has reached nearly 50 lb. absolute, indicating a large waste of energy.

85. Fig. 14 shows the combined air cards, taken from both the head-end and crank-end of a two-stage Nordberg compressor. A comparison of these cards with the one shown in Fig. 13

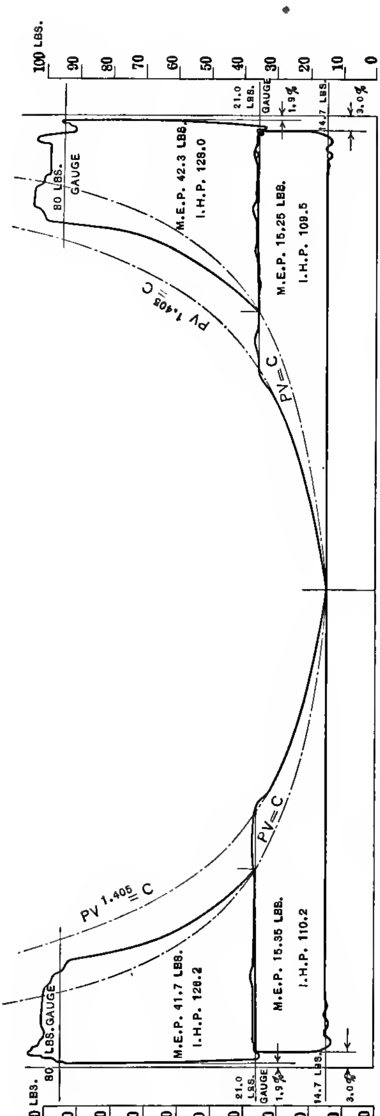


FIG. 14.—Combined Air Cards of a Two-stage Nordberg Compressor.

reveals far more perfect conditions. The in-take line of the low-pressure cylinder closely follows the atmospheric line, showing unrestricted in-take areas. The compression line of the high-pressure cylinder begins at the intersection of the isothermal compression line with the delivery line of the low-pressure cylinder, showing perfect inter-cooling; and the fact that the discharge line of the low-pressure cylinder is practically identical with the in-take line of the high-pressure cylinder and is nearly a straight line, shows satisfactory condition of valves and proper proportioning of the cylinders for the prevailing ratio of compression.

Low-pressure air cylinder	29×42 in.
High-pressure air cylinder	19×42 in.
Piston speed	364 ft. per minute.
Piston displacement per stroke	15.84 cu. ft.
Pressure of inlet air	14.7 lb. absolute.
Discharge pressure	80.0 gage.
Volumetric efficiency (from card)	98.7 per cent.
Actual free air per minute (from card)	1611 cu. ft.
Air horse-power (from card)	237

The theoretical horse-power required for single-stage compression of the same quantity of air is 256, showing a saving of power over single-stage compression amounting to 8 per cent.

CHAPTER IX

COOLING WATER REQUIRED IN COMPRESSION; EFFICIENCY OF COMPRESSOR PLANT; AIR-COMPRESSOR EXPLOSIONS

86. Amount of Cooling Water Required in Air Compressors.

Let W = weight of required water in pounds per unit of time.
w = weight of air in pounds to be cooled per unit of time.
t = initial temperature of free air in degrees Fahr.
= initial temperature of cooling water in degrees Fahr.
t_1 = final temperature of compressed air in degrees Fahr.
s = specific heat of air.

We have seen that the amount of heat required to raise the temperature of,1 lb. of water 1° Fahr. is 1 B.T.U. Accordingly, the number of pounds of water required to abstract a quantity of heat from any substance without raising the temperature of the water more than 1° Fahr. is equal to the number of B.T.U.'s to be abstracted.

Now, if "w" pounds of air have been heated during compression from an initial temperature $t°$ to a final temperature $t_1°$ Fahr., the rise in temperature is $(t_1-t)°$ Fahr.

In order to cool this quantity of air to initial temperature, we must abstract from it an amount of heat, which, expressed in B.T.U.'s, is

$$\text{B.T.U.'s} = w(t_1-t)s \tag{1}$$

This also represents the number of pounds of cooling water required, having the same initial temperature as the in-take air. Therefore

$$W = w(t_1-t)s \tag{2}$$

Since in compressor practice the air is cooled under constant pressure, the specific heat is from Article 8.

$$s = 0.2375$$

Example.—How many gallons of water per minute are required to cool to initial temperature 800 cu. ft. of free air per minute, compressed

to 80 lb. gage? Initial temperature of water and free air to be 60° Fahr.; temperature of water when leaving cooling devices to be not more than 61° Fahr.

One pound of water = 0.12 gal.

From column 9, Table V, we find the final temperature of air, compressed adiabatically in two stages to 80 lb. gage = 224° Fahr., which is an increase of $(224-60)=164°$.

From Table I, 800 cu. ft. of free air at 60° Fahr. weigh $800\times 0.0764=$ 61.12 lb.

Therefore B.T.U.'s to be abstracted from air $= 61.12\times 164\times 0.2375=$ 2380 and water required per minute $= 2380\times 0.12=290$ gal.

In practice it is usually deemed satisfactory to allow an increase of temperature in the water, amounting to from 10 to 25 degrees, thereby reducing the quantity of cooling water required. If the initial temperature of the water is 60° Fahr. when entering and 70° Fahr. when leaving the cooling devices, then every pound of water has absorbed 10 B.T.U.'s from the heated air and the number of pounds of water required in this case will be only one-tenth of the quantity stated above, that is $\frac{290}{10} = 30$ gal. per minute (nearly).

In actual compression, considerable radiation is going on and the final temperature of the compressed air is less than the theoretical temperature taken from the table. It is therefore safe to divide the theoretical quantity of water as found from equation (2) by two or even three.

For the case assumed, the minimum quantity of cooling water required would therefore be:

$$\frac{30}{3}=10 \text{ gal. per minute.}$$

When water is scarce and has to be used over and over again, it becomes necessary to cool it by artificial means to initial temperature before returning it to the cooling devices of the compressor. The greater its temperature when leaving the compressor, the slower and the more expensive will be the artificial cooling which is usually accomplished in so-called cooling-towers. Hence the practical limit of permissible increase in temperature, which is, as stated above, from 10 to 15 degrees above initial temperature.

EFFICIENCY OF A PLANT FOR THE PRODUCTION OF COMPRESSED AIR

87. Compressor efficiency is a term which is used rather loosely. Usually it is meant to designate the ratio between the theoretical power required to compress and deliver a certain

quantity of free air and the actual power expended. This is the mechanical efficiency of a compressor, strictly speaking. Volumetric efficiency is explained under Article 53.

What is important to know in the end is the efficiency of the complete plant for the production of compressed air, including the compressor itself with all its accessories such as air in-takes, inter-coolers, after-coolers, receivers, and so forth.

We wish to know the power value of a certain quantity of compressed air after it has left the compressor and has cooled down to initial temperature. All energy expended beyond that value is lost for useful purposes. This loss is chargeable to the installation for the production of compressed air and its total amount measures the lack of efficiency of the installation.

To compute the efficiency, we must take into account all losses that occur from the moment the air is taken into the cylinder of the compressor until it is delivered into the pipe line at practically initial temperature.

These losses have been frequently alluded to in previous articles. They are as follows:

1. Losses are due to initial temperature of the in-take air. It has been pointed out that under ordinary circumstances air will, after compression and before use, assume the temperature of natural objects. For compressor computations this is usually taken as 60° Fahr.

We will assume that we wish to produce a volume of air compressed to 70 lb. gage which, after having cooled down to 60° Fahr., is the equivalent of 500 cu. ft. of free air per minute. If the in-take air has a temperature of 60° Fahr., the amount of air to be compressed is, of course, 500 cu. ft. per minute. To compress adiabatically in one stage 500 cu. ft. of free air per minute to 70 lb. gage, and deliver it into the receiver, requires (from column 4, Table V) theoretically 74.00 h.p.

If the temperature of the in-take air had been 100° Fahr. and 500 cu. ft. of it were cooled down under constant pressure to 60 degrees, they would occupy a volume of:

$$V_1 = V\frac{T_1}{T}.$$

$$V_1 = \frac{500 \times (60+461)}{(100+461)} = 464 \text{ cu. ft.,}$$

which is less than the required volume.

Therefore, in order to have in the end a volume of compressed air which when cooled to 60° Fahr. is the equivalent of 500 cu. ft. of free air, we should have compressed more than 500 cu. ft. of the 100 degree air. The volume to be compressed in this case is:

$$V=\frac{500\times(100+461)}{(60+461)}=538 \text{ cu. ft.}$$

To compress adiabatically in one stage 538 cu. ft. of free air per minute to 70 lb. gage, and deliver it into the receiver, requires (from column 4, Table V) theoretically 79.62 h.p. This is an increase of 7 per cent. in the required power due to an increase of 40 degrees in the temperature of the in-take air, or 1 per cent. for every 6 degrees.

This points to the advantage of bringing cool air to the machine. Neglect to do so will result in power loss as pointed out, which must be charged to the compressor plant as a whole.

2. Losses are due to the in-take air being heated as it passes in very thin streams over the valve surfaces which have been heated by the air under compression. These losses are difficult to ascertain, since an indicator card gives no hint whatever of their occurrence. Indicators record pressure only, not temperatures. These losses add to those stated under (1), and may be considerable.

3. Losses are due to imperfect valves. Poppet valves being operated by strong springs are liable to throttle the inlet air, the effect of which is scarcely noted on ordinary indicator cards. Nevertheless, such throttling results in the creation of a partial vacuum which cuts down the capacity as pointed out under Article 53 and being a drag on the machine consumes more power in addition to power required for the extra number of revolutions needed to make up for capacity loss.

4. Losses are due to clearance. They have been discussed in Article 52. The ultimate effect of clearance is the delivery, into the receiver, of a volume of air smaller than that which has actually been compressed at each stroke of the piston.

5. Losses are due to the generation of heat during compression which is afterward dissipated and completely lost for useful work. These are by far the most serious losses incident to air compression, and have therefore received close attention by designers and builders of air compressors. All attempts have

been directed toward the accomplishment of isothermal compression by the introduction of water-jackets and of inter-coolers in stage compressors. At the present state of the art, these losses are unavoidable. But by judicious selection of compressors provided with adequate cooling devices they can be reduced to a minimum.

6. Losses are due to imperfect design, carelessness in handling, neglect in properly lubricating, stopping leaks, and removing worn-out parts of the compressor.

Example.—Let the piston displacement of the air cylinders of a single-stage compressor be 10 cu. ft. and the temperature of the free air taken into this cylinder be 60° Fahr. In passing over the heated inlet valves and coming in contact with the heated cylinder walls, its temperature will rise, let us say, to 70° Fahr. We have no means at present to measure this accurately. The theoretical power required to compress in one stage adiabatically 10 cu. ft. of free air to 70 lb. gage and deliver it into the receiver is:

$$W_n = \frac{144\ nP_1V_1}{n-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ ft.-lb.}$$

$$= \frac{144\times 1.406\times 14.7\times 10}{0.406}\left[\left(\frac{84.7}{14.7}\right)^{0.29} - 1\right] = 48{,}400 \text{ ft.-lb.}$$

To this we will add 15 per cent. for friction etc., which gives

$$W_n = 55{,}600 \text{ ft.-lb.} \tag{1}$$

Neglecting jacket cooling and radiation, the temperature of the compressed air will be:

$$T_1 = T\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} = (70+461)\times\left(\frac{84.7}{14.7}\right)^{0.29} = 882 \text{ degrees absolute.}$$

The volume V_2 into which the air has been compressed will be:

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} = 10\left(\frac{14.7}{84.7}\right)^{0.71} = 2.88 \text{ cu. ft.}$$

If the clearance of the compressor is 2 per cent., the actual volume of compressed air delivered into the receiver is not V_2 but:

$$V_3 = 2.88 - \frac{10}{100}\times 2 = 2.68 \text{ cu. ft.}$$

It is true that the expanding clearance air helps to compress air on the return stroke of the piston, that is, the value of W_n as stated in equa-

tion (1) will be somewhat less. On the other hand, this clearance air having a temperature of 882 degrees absolute, in mingling with the incoming free air will raise its temperature somewhat beyond that which we have allowed for heating during contact with the heated inlet valves, so, for the sake of demonstration we shall neglect the gain in required power for compression.

Now, the pressure of this volume V_3 of air, having a temperature of 882 degrees absolute, in cooling down to a temperature of 60° Fahr. under constant volume, will decrease to a pressure P_3 which we deduce from formula in Article 19:

$$P_3 = P_2 \frac{T_3}{T_2}$$

$$P_3 = (70+14.7)\left(\frac{60+461}{882}\right) = 50 \text{ lb. absolute.}$$

We have then in the end a volume of 2.68 cu. ft. of air at an absolute pressure of 50 lb. and a temperature of 60° Fahr., which, if allowed to expand adiabatically to initial pressure (14.7 lb. absolute) is theoretically capable of performing an amount of work, which we deduce from formula (1), Article 110.

$$W_n = \frac{144\, nP_3V_3}{n-1}\left[1-\left(\frac{P_1}{P_3}\right)^{\frac{n-1}{n}}\right]$$

$$= \frac{144\times 1.406\times 50\times 2.68}{0.406}\left[1-\left(\frac{14.7}{50}\right)^{0.29}\right] = 20{,}000 \text{ ft.-lb.}$$

This is the power value of the compressed air which we started out to determine.

The efficiency of our compressor plant, that is, of the complete installation for the production of compressed air is in the case under consideration:

$$\frac{20{,}000}{55{,}600} = 0.36 = 36 \text{ per cent.}$$

Under favorable conditions the actual efficiency may be greater, due to the fact that the final temperature of the compressed air will be less than 882 degrees absolute because of jacket cooling and radiation. In many plants the efficiency is considerably less.

For stage compressors the theoretical efficiency is higher than for single-stage compressors, because the work of compression and delivery is less. But a stage compressor is a more expensive machine and only warranted when the reduction in

operating costs more than compensates for the original investment.

Practical stage-compressor tests frequently show an excess of power consumed, rather than a saving over single-stage compression. (See Chapter VIII on Indicator Cards.)

AIR-COMPRESSOR EXPLOSIONS

88. In general, an explosion is due to quick combustion, followed by the generation of a large volume of gas, which, if confined, suddenly increases the pressure against the enclosing walls beyond their strength of resistance.

If an explosion occurs in an air compressor, we must trace the cause to the presence in the air cylinder of:

1. A combustible substance in a finely divided state so as to permit practically instantaneous ignition of the whole mass and therefore the sudden generation of a large volume of gas.

2. Of air or oxygen in the proper proportion to completely oxidize the combustible, so as to form an explosive mixture.

3. A temperature high enough to ignite the mixture.

The absence of any one of the above three conditions will make an explosion impossible. This points to the preventive measures enumerated below.

1. Combustibles: The only combustible substance purposely introduced into the air cylinder of a compressor is the lubricating oil. Incidentally, whatever combustible substance may be contained in the atmosphere will be drawn into the cylinder during the suction stroke. If consisting of fine coal dust, for instance, the danger of an explosion will be materially increased.

One of the ingredients of a lubricating oil is carbon, which will separate out from the oil under the influence of heat and pressure and, if permitted, will eventually accumulate in such quantities as to make an explosive mixture with the air.

The requirement that the combustible must be finely divided is satisfied by the vapors of the oil, given off under high temperatures. When it is considered, however, that the vapors thus formed are expelled from the air cylinder with each stroke, only the most reckless use of lubricating oil would furnish enough vapors during one stroke of the piston to form an explosive mixture with the air. Thus it becomes highly improbable that the presence of such vapors alone ever causes an explosion. It

is nevertheless possible that in an air cylinder with leaky valves and piston, enough of these vapors may eventually accumulate to at least assist in an explosion.

It is more probable that carbonaceous matter in a finely divided or porous state is the chief cause of an explosion, when present in the proper proportion and exposed to abnormal temperatures.

2. The proportion of combustible matter and air or oxygen, required to form an explosive mixture, is dependent on the nature of the combustible. We are fairly well acquainted with these proportions when the mixture takes place under atmospheric pressure. What they are under the high pressures prevailing in the cylinder of an air compressor is a question which can only be answered when more reliable data become available.

3. The temperature required to ignite an explosive mixture under atmospheric pressure we also know fairly well. Finely divided carbon, for instance, ignites at a temperature of 600° Fahr. Under high pressure, it is quite possible that ignition takes place at a temperature considerably below that.

However conducive oil may be to an explosion, its use as a lubricant, and therefore the accumulation of carbonaceous matter and the formation of vapors, cannot be avoided altogether. But by proper care and perfect cooling devices, the temperature can be kept within safe limits. To do this, we must know the conditions which cause abnormal temperatures. Chief among them are the following:

a. When air is taken into the cylinder from a hot engine room or from the neighborhood of a boiler room. Other evil effects of such conditions have already been pointed out under Article 87.

Air at a temperature of 150° Fahr. and compressed in one stage to, say, 75 lb. gage, would have a final temperature of

$$T_1 = T\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} = (150+461)\left(\frac{75+14.7}{14.7}\right)^{0.29} = 1030^\circ \text{ absolute}$$
$$= 569^\circ \text{ Fahr.}$$

which even if reduced somewhat by water-cooling would come dangerously close to or exceed the ignition point of any explosive substance.

b. When the maximum pressure for which the compressor is built is willfully or accidentally exceeded. The final temperature is thus raised to a point beyond the efficiency of the cooling ap-

pliances and will ultimately reach the ignition point of an explosive mixture.

c. When the pressure is exceeded by reason of a gradual accumulation of carbonaceous matter of the oil in the discharge valves. This contracts the passage and requires higher pressure for the delivery of a certain amount of air in a given time, thus raising the temperature.

d. When valves are not kept clean and pistons are permitted to wear loose, which causes leakage of compressed air into the in-take side of the cylinder. This is probably the most frequent source of excessive temperature. The leakage air expands to atmospheric pressure without doing work, hence loses none of its heat (see Article 118), and in mixing with the incoming free air it raises the temperature of the mixture which is to be compressed on the following stroke to a dangerous degree.

The dangerous effect of leakage in the air cylinders of a compressor is shown in the following example:

Example.—Let W_0 = weight of a given quantity of atmospheric air, occupying a volume equal to the piston displacement.

= unity (assumed) $= 1 = W_1 + W_2$.

W_1 = weight of atmospheric air which enters the cylinders at each stroke at an absolute temperature T_1.

$= 1 - W_2$.

W_2 = weight of leakage air, having an absolute temperature T_2 and which expands to atmospheric pressure upon entering the in-take side of the cylinder.

P_0 = absolute pressure of atmospheric air in pounds per square inch.

P_2 = absolute pressure of compressed air in pounds per square inch.

c = specific heat of air.

n = 1.406.

It is evident that the total quantity of heat in the mixture W_o must be equal to the sum of the heat quantities contained respectively in W_1 and in W_2 before mixing. If the absolute temperature of W_o has become T_o after the mixing of the leakage with the atmospheric air, then:

$$\text{Total heat in } W_0 = W_oT_oc$$
$$\text{Total heat in } W_1 = W_1T_1c$$
$$\text{Total heat in } W_2 = W_2T_2c$$

and $$W_oT_oc = c(W_1T_1 + W_2T_2)$$

Remembering that $W_o = 1$ and $W_1 = 1 - W_2$

We find $$T_o' = T_1(1 - W_2) + T_2W_2 \tag{1}$$

If the temperature before compression is T_o, then the temperature T_2 after compression to an absolute pressure P_2 according to equation (11), Article 41, is:

$$T_2 = T_o\left(\frac{P_2}{P_o}\right)^{\frac{n-1}{n}} \tag{2}$$

Let the temperature of the in-take air be 60° Fahr.

then $T_1 = 60 + 461 = 521$ degrees absolute.

Let leakage air equal 15 per cent. of W_o

then $$W_2 = \frac{W_o}{100} \times 15 = \frac{1}{100} \times 15 = 0.15$$

Let the air be compressed to 85 lb. gage.

then $$P_2 = 85 + 14.7 = 99.7$$

If we assume the compressor to have made a number of strokes before leakage occurs, then the temperature of the compressed air will be 447° Fahr. (Col. 8, Table III). If leakage now begins at a rate of 15 per cent. we have T_o from equation (1):

$T_o = (60+461)(1-0.15)+0.15(447+461) = 579$ degrees absolute.

If we compress this air to 85 lb. gage, its final temperature will be from equation (2):

$T_2 = 579 \times 6.782^{0.29} = 1005$ degrees absolute $= 544°$ Fahr.

Some of this compressed air leaks into the in-take side of the cylinder where we get an air mixture of the temperature:

$T_o = (60+461)(1-0.15)+0.15(544+461) = 595$ degrees absolute

and this air compressed to 85 lb. gage will have a temperature

$T^2 = 595 \times 6.782^{0.29} = 1037$ degrees absolute $= 576°$ Fahr., etc.

This shows that the temperature in the cylinder is increasing with every stroke of the piston and in spite of the cooling devices will soon reach a point at which in the presence of combustible material and inflammable vapors an explosion is likely to occur.

The danger increases when the compressor is running at slow speed. Leakage is a constant quantity per unit of time. Hence in a slow-speed machine the percentage of leakage air per stroke will increase and that of the cool in-take air will decrease. As a consequence the temperature of the mixture to be compressed on the next stroke will be proportionally higher, thereby increasing the danger of an explosion.

89. Compressors, using throttling devices for the regulation of intermittent demand, may under certain conditions develop dangerous temperatures. By throttling the inlet, the initial pressure is lowered, while the ratio of compression is increased. As a result, the final temperature will be considerably higher than under normal conditions, so high as to ignite any explosive substance that may be present in the cylinder.

Example.—Let us assume that the output of a single-stage compressor, working against 60 lb. pressure is to be reduced by throttling to one-fourth of its full capacity. If atmospheric pressure is 14.7 lb. per square inch, the pressure of the in-take air will become:

$$\frac{14.7}{4} = 3.7 \text{ lb. per square inch,}$$

and the total ratio of compression:

$$\frac{P_1}{P} = \frac{60+14.7}{3.7} = 20.2$$

If the initial temperature of the air is 70° Fahr., the final temperature, from equation (11), Article 41, will be:

$$T_1 = T\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} = (70+461)\,20.2^{0.29} = 1270 \text{ degrees absolute, } 809° \text{ Fahr.}$$

Although radiation and jacket-cooling will considerably reduce this temperature, the figures nevertheless indicate the danger connected with compression of rarefied air such as obtains in compressors regulating the output by throttling the in-take.

90. Prevention of Compressor Explosions.—Having pointed out the most likely causes of an explosion in a compressor, the means of prevention become self-evident. They consist first in guarding against accumulation of explosive substances in the air cylinder, and second in keeping down temperatures below the danger point.

The first is accomplished by using a lubricant which does not precipitate its carbon contents at normal temperatures and by using it sparingly. Soapy water fed at intervals to the cylinder is good practice. Inspect the valves frequently and remove all accumulations of foreign substances.

Abnormal temperatures are prevented by water-jacketing and inter-coolers, if they are of proper design and magnitude. Leaks in valves and piston, as pointed out, prevent the most perfect cooling devices from doing their duty. They should be stopped as soon as discovered.

PART II

THE TRANSMISSION OF COMPRESSED AIR

CHAPTER X

TRANSMISSION OF COMPRESSED AIR

91. Compressed air, before being used in so-called air engines, has to be conveyed from the compressor room to the points of use in iron pipes of various dimensions. The question then arises: What should be these dimensions so as to satisfy the demand for compressed air at the discharge end of the pipe line?

The solution of this problem requires, besides a knowledge of the behavior of compressed air flowing through a pipe line, a careful study of local conditions and a close comparison of first cost of installation with the ultimate operating expenses.

92. The laws governing the flow of compressed air in iron pipes are far more elusive and complex than those for water, owing to the fact that compressed air is not a stable substance like water but changes its pressure, density, volume, velocity and friction at every point of the pipe line.

Attempts to express these laws in simple formulas, which would hold good in all cases, seem therefore quite hopeless. The best that can be said of any of the formulas employed by engineers, is that the numerical results obtained from them will in most cases correspond only approximately with those obtained from actual practice. They must therefore be used with a great amount of caution.

Before referring to these formulas, it is well to study the behavior of compressed air during its passage through a pipe line. We have seen that in order to cause air to flow from one point to another, there must be a difference of pressure in the air at those two points. That is, if we need air at a pressure of 80 lb. at a certain distance from the compressor, the pressure of the air as it leaves the compressor or receiver must be greater than 80 lb.

This difference of pressure which is necessary to make the air flow, represents energy which does useful work, that is, it conveys the air from one place to another. It is therefore not a loss in the true meaning of the word. But air in its passage through pipes is subject to friction. To overcome this friction, energy in the

form of pressure is required, additional to that needed for keeping the air in motion. To produce this extra pressure in the air, power is consumed in the compressor which is an actual loss as there can be no useful return for it.

Friction reduces the pressure of the air, and since the temperature in the pipe can be assumed to be constant, this reduction of pressure increases the volume of the air. To force an increased volume of air through a pipe of the same diameter during the same time means increased velocity and this in its turn requires an additional power in the form of pressure.

In general, the transmission of air in pipes entails both a loss of pressure and a loss of power.

93. Loss of pressure or head is the difference of pressure in the air between the in-take and the discharge end of the pipe line. This loss as has been mentioned is due:

1. To pressure consumed in causing the air to flow from one end of the pipe line to the other. It is as explained, not a loss, strictly speaking.

2. To pressure consumed in overcoming friction and increasing the velocity. This loss is unavoidable but can be minimized by intelligent design of the pipe line.

3. To leakage which causes the air remaining in the pipe to expand and therefore lose pressure. This loss is avoidable and in a well-constructed pipe line should be practically nil.

4. To difference in elevation between the compressor room and the points of use. (See Article 97.)

94. Loss of Power.—The ultimate loss of power which is chargeable to transmission, is the difference between the amount of power, residing in the compressed air when it enters the pipe line and that which is available at the discharge end of the pipe line. (See also Articles 97 and 102.)

Since the amount of work required to compress air and the amount of work which compressed air is capable of performing, depends on pressure as well as on volume and since loss of pressure increases the volume (the temperature remaining the same), it follows that the loss of power is not in direct proportion to the decrease in pressure but is partly compensated by the resultant increase in volume.

For instance, if air enters a pipe at 100 lb. and is discharged at 80 lb. gage, there is a *loss in pressure* of 20 per cent. From equation (1), Article 110 the theoretical work which 1 cu. ft. of

compressed air is capable of performing in expanding adiabatically from 100 lb. gage to atmospheric pressure is 25,740 ft.-lb. This represents the potential energy of 1 cu. ft. of air when it enters the pipe line at 100 lb. gage. This cubic foot of compressed air when its pressure is reduced to 80 lb. at the end of the pipe line has expanded into a volume of 1.211 cu. ft., which at a pressure of 80 lb. is capable of doing work to the amount of 24,000 ft.-lb. Hence the *loss of power* in this case amounts to only 7 per cent. as against a pressure loss of 20 per cent.

Loss of pressure in air transmission must therefore not be confounded with loss of power. Both losses in a well-proportioned pipe line are usually small compared with other losses in the production and use of compressed air, such as result, for instance, from the heat produced during compression which is subsequently lost in transmission.

CHAPTER XI

DIMENSIONS OF PIPE-LINES FOR CONVEYING COMPRESSED AIR

95. From what has been said, it is evident that no uniform rule can be followed in deciding on the proper dimensions of pipes for air transmission. Almost any degree of transmission efficiency can be obtained by using pipes of large diameter. This, however, may result in extravagant first cost. On the other hand, an unwise economy in first expenditure may reduce the efficiency of the whole system to a point where the cost of operation will more than offset the original saving in cost of installation.

A proper design of pipe line must therefore take into consideration not only first cost of pipe and interest thereon but also the subsequent operating costs. The latter will vary with the size of the compressor, the pipe line, and with local conditions such as cost of transportation, fuel, labor, etc., in the part of the country where the plant is to be erected. The problem must be solved for each individual installation.

In order to make comparisons, however, we must have means of calculating, at least approximately, the minimum dimensions of a pipe which will fill the requirements of a certain installation. Or, if the size of the pipe is given, we must be able to calculate the work which the compressor must perform in order that the available power at the discharge end of the pipe line will be a certain quantity. Or, if a certain compressor and pipe line are on hand, we must be able to calculate the power that will be available at the discharge end of the pipe line before we buy and install our air engines.

The conditions that affect pressure, volume, density, temperature, velocity and friction of the air passing through a pipe line, are so numerous that the formulas proposed by authorities cannot be expected to give more than safe approximate results for pipe line dimensions. Such results should be considered the maxima or minima in each case and ample allowances should be made for undetermined contingencies.

The formulas selected for this treatise are those proposed by Johnson-Rix.

96. Formulas for Pipe Line Computations.—

Let P_1 and P_2 = absolute pressures at intake and discharge terminals of pipe, in pounds per square inch.

V = volume of free-air equivalent in cubic feet per minute, passing through pipe.

L = length of pipe line in feet.

D = diameter of pipe in inches.

Then
$$P_1^2 - P_2^2 = \frac{V^2L}{2000D^5} \qquad (1)$$

$$D = \sqrt[5]{\frac{V^2L}{2000(P_1^2 - P_2^2)}} \qquad (2)$$

$$V = \sqrt{\frac{2000D^5(P_1^2 - P_2^2)}{L}} \qquad (3)$$

$$L = \frac{2000D^5(P_1^2 - P_2^2)}{V^2} \qquad (4)$$

$$P_1 = \sqrt{\frac{V^2L}{2000D^5} + P_2^2} \qquad (5)$$

$$P_2 = \sqrt{P_1^2 - \frac{V^2L}{2000D^5}} \qquad (6)$$

Example 1.—What should be the diameter of a pipe line, 1200 ft. long that will transmit the equivalent of 4000 cu. ft. of free air per minute with a pressure loss of not more than 8 lb. Initial pressure = 100 lb. gage. Atmospheric pressure = 14.7 lb.

From equation (2)
$$D = \sqrt[5]{\frac{4000^2 \times 1200}{2000(114.7^2 - 106.7^2)}}$$

whence $D = 5.58$, say 6 in.

Example 2.—What is the free air equivalent in cu. ft. per minute that can be transmitted through an 8-in. pipe line 11,000 ft. long so that the pressure loss is not more than 5 lb. Initial pressure = 80 lb. gage. Atmospheric pressure at locality selected = 12.02 lb.

From equation (3)
$$V = \sqrt{\frac{2000 \times 8^5(92.02^2 - 87.02^2)}{11{,}000}}$$

whence $V = 2300$ cu. ft. of free air per min.

Example 3.—What must be the initial pressure of an equivalent of 500 cu. ft. of free air per minute so that at the end of a 2-in. pipe line,

300 ft. long it has a pressure of 80 lb. gage. Atmospheric pressure at locality selected = 11.30 lb.

From equation (5) $$P_1=\sqrt{\frac{500^2\times 300}{2000\times 2^5}+91.30^2}$$

whence $P_1=97.5$ lb. absolute

$=86.2$ lb. gage

Example 4.—What will be the terminal pressure of an equivalent of 500 cu. ft. of free air per minute at the discharge end of a 5-in. pipe line, 2500 ft. long, when initial pressure = 90 lb. gage. Atmospheric pressure at locality selected = 12.02 lb.

From equation (6) $$P_2=\sqrt{102.02^2-\frac{500^2\times 2500}{2000\times 5^5}}$$

whence $P_2=101.02$ lb. absolute

$=89$ lb. gage

Example 5.—What is the permissible length of a 6-in. pipe line that will transmit the equivalent of 2000 cu. ft. of free air per minute with a pressure loss of not more than 10 lb. Initial pressure = 90 lb. gage. Atmospheric pressure = 14.7 lb.

From equation (4) $$L=\frac{2000\times 6^5(104.7^2-94.7^2)}{2000^2}$$

whence $L=7750$ ft.

97. Effect of Altitude on the Transmission of Compressed Air.—The formulas given under Article 96 do not take into consideration any difference of elevation between the in-take and the discharge end of the pipe line, but assume that the two terminals of the pipe line are practically at the same elevation. In compressed-air installation, it happens frequently that the engines using the compressed air are located at a considerable elevation above the compressor in which case proper allowance must be made for the loss of pressure due to this fact.

Example.—At a mine located 1500 ft. above the compressor house, a free air equivalent of 4000 cu. ft. per minute is required at a gage pressure of 80 lb. The length of the pipe line is 2600 ft. Elevation at compressor is 3000 ft. above sea level.

(*a*) If a loss of 6 lb. is allowed for pipe friction, what must be the gage pressure of the air when leaving the compressor?

(*b*) What should be the diameter of the pipe?

Solution (*a*) From Table VI atmosph. press. at compressor = 13.16 lb. From equation (4), Art. 4, atmosph. press. at mine

$$\log P_{4500} = \log 13.16 - \frac{1500}{122.4(60+461)} \text{(assuming temp.} = 60°\text{F.)}$$

Whence $P_{4500} = 12.32$ lb.

Hence, absolute pressure of compressed air at mine

$P_2 = 80 + 12.32 = 92.32$ lb. abs. at discharge terminal of pipe line.

The corresponding pressure at the intake of the pipe is found from the same formula:

$$\log P_1{}^1 = \log 92.32 + \frac{1500}{122.4(60+461)}$$

Whence $P_1{}^1 = 98.65$ lb. absolute.

Adding to this the 6 lb. allowed for pipe friction, it follows that the compressed air leaving the compressor must have a pressure of

$$P_2 = 104.65 \text{ abs. or } 104.65 - 13.16 = 91.49 \text{ lb. gage}$$

in order that the pressure at the mine may be 80 lb. gage. Obviously on account of difference in altitude alone, an extra pressure of $98.65 - 92.32 = 6.33$ lb. is required.

Solution (*b*). Introducing the respective values in equation (2), Art. 96, we get diameter of pipe

$$D = \sqrt[5]{\frac{4000^2 \times 2600}{2000(104.65^2 - 98.65^2)}}$$

Whence $D = 6$ in.

98. Dimensions of Branch Pipes.—In selecting the dimensions for branch pipes to carry compressed air, it must be borne in mind that the carrying capacity of a pipe is not directly proportional to the cross-section of the pipe. Under the same conditions of length and head a 3-in. pipe, for instance, will carry only 16 per cent. of the volume which a 6-in. pipe can carry. Therefore if a 6-in. main is to be divided into two branches, two 3-in. pipes would not do the work, neither would the combined capacities of a 4- and 5-in. pipe be sufficient. This will be seen from Table IX.

Going to the column showing the capacities for a 6-in. pipe and following this column down to the figure opposite 4 in., we find that the capacity of a 4-in. pipe is only 35 per cent. of the capacity of a 6-in. pipe and for the 5-in. pipe we find it to be 63 per cent. of the 6-in. pipe capacity. The sum of the 4- and 5-in. pipe capacities is therefore only 98 per cent. of the 6-in. pipe capacity, resulting in a slight additional friction from the point

of diversion of the branches. If the branches were made of 4 1/2- and 5-in. pipes, the percentages would be 47 and 63 respectively, their sum 110 per cent., resulting in a slight decrease of friction and therefore an easier flow beyond the diversion of the branches.

99. Effects of Elbows and Bends on the Flow of Air in Pipes.—Bends and elbows in a pipe line have the effect of increasing the friction of the air and thus reduce the pressure. In table below which is taken from the Trade Catalogue of the Norwalk Iron Works Co., is given the length of pipe in terms of diameters which will produce the same frictional effect as an elbow having a certain radius.

For instance, the frictional resistance in a 6-in. pipe line 500 ft. long containing five elbows with a radius of 18 in. or three diameters each, would be the same as that produced by a straight pipe line $500+\frac{5\times 8.24\times 6}{12}=520$ ft. long.

From the table the beneficial effect of a gradual curve in comparison with a short sharp turn, is quite evident.

Radius of elbow 5 diameters.	Equivalent length of straight pipe 7.85 diameters.
Radius of elbow 3 diameters.	Equivalent length of straight pipe 8.24 diameters.
Radius of elbow 2 diameters.	Equivalent length of straight pipe 9.03 diameters.
Radius of elbow 1 1/2 diameters.	Equivalent length of straight pipe 10.36 diameters.
Radius of elbow 1 1/4 diameters.	Equivalent length of straight pipe 12.72 diameters.
Radius of elbow 1 diameter.	Equivalent length of straight pipe 17.51 diameters.
Radius of elbow 3/4 diameter.	Equivalent length of straight pipe 35.09 diameters.
Radius of elbow 1/2 diameter.	Equivalent length of straight pipe 121.20 diameters.

100. Velocity of Compressed Air Flowing Through a Pipe Line.—

Let V_c = volume of compressed air in cubic feet per minute.

v = velocity in feet per minute.

A = area of pipe section in square feet.

then $$v=\frac{V_c}{A} \text{ ft. per minute.} \qquad (1)$$

If in Example (2), Art. 96, the average pressure of the air is taken as $\frac{92.02+87.02}{2}=89.52$ lb. abs.

Then $\frac{V_c}{V}=\frac{12.02}{89.52}$ Whence $V_c=\frac{2300\times 12.02}{89.52}=310$ cu. ft.

The sectional area of an 8-in. pipe is 0.35 sq. ft.

Hence $v=\frac{310}{0.35}=885$ ft. per min. or 14.8 ft. per second.

Although pressure losses in a pipe line are obviously a function of the volume of air transmitted per minute and therefore of velocity, as well as of the length, the diameter and roughness of the pipe line, the individual effect of these factors has not as yet been definitely determined. It is generally conceded, however, that for economical transmission the actual velocity of the air in a line constructed for long continuous operation should not materially exceed 30 ft. per second. If either the diameter of the pipe or the volume of the air to be transmitted are fixed, the velocity is decreased in the first case by limiting the volume of air, and in the second case by increasing the diameter of the pipe.

100A. The Planning of a Transmission Line.—It has been shown that the transmission of compressed air involves numerous losses depending on the size, length and roughness of the pipe, on the number of bends and elbows, on the volume of air to be transmitted and the initial pressure with which it enters the pipe line.

The intelligent planning of the pipe line should therefore aim to minimize these losses as much as practicable by transmitting the air to the points of use over the shortest possible route, requiring the least number of elbows, joints and other obstructions, and by selecting this route so that the pipe is accessible at all points. This is desirable to permit frequent inspection and stoppage of leaks which, as a rule, are the chief cause of extreme pressure losses.

It usually pays to do a certain amount of grading and excavating in order to eliminate sudden changes in both the horizontal and vertical alignments of the pipe line.

If air is to be transmitted through a network of pipes, not

only the diameter of the air-main but the diameters of all the branch pipes should be calculated with the highest degree of accuracy.

This is accomplished by applying the formulas in Art. 96 to both types of pipes.

Example.—A compressor furnishes to a mine air that has a pressure of 100 lb. absolute at the intake terminal of a 6-in. air main. The free-air equivalent is 3000 cu. ft. per min. At a point, 2600 ft. from the compressor (measured along the pipe line) a 1200-ft. branch line is to be taken off to supply air at 87 lb. absolute to six stoping drills, whose combined free-air consumption is 300 cu. ft. per min. What should be the diameter of the branch pipe?

For the solution of this problem it is necessary to know the normal air pressure at the point of take-off. In an existing pipe line the pressure may be obtained by attaching a pressure gage to the air main at that point. But if the branch pipes are to be planned simultaneously with the main pipe, the expected pressure at the point of take-off must be calculated by equation (6) Art. 96.

$$P_2 = \sqrt{P_1^2 - \frac{V^2L}{2000D^5}}$$

or

$$P_2 = \sqrt{100^2 - \frac{3000^2 \times 2600}{2000 \times 6^5}} = 92 \text{ lb. abs.}$$

The minimum diameter of the branch pipe will be from equation (2), Art. 96

$$D = \sqrt[5]{\frac{V^2L}{2000(P_1^2 - P_2^2)}}$$

or

$$D = \sqrt[5]{\frac{300^2 \times 1200}{2000(92^2 - 87^2)}} = 2.5 \text{ in.}$$

A pipe smaller than that would supply neither the required quantity of air nor the required pressure.

PIPE-LINE EFFICIENCY

101. The efficiency of a pipe line is the ratio between the available energy for doing useful work residing in the air at the discharge end, and that which is available at the in-take end of the line.[1] In computing this ratio the temperature of the compressed air must be assumed to be the same at both terminals of the pipe line. For, whatever the dimensions of the pipe, the compressed

[1] See also Article 102.

air at the discharge end will be practically at outside temperature, that is, at the temperature of the free air taken into the compressor cylinder. Any extra heat left in the compressed air when it enters the pipe line should be charged to the efficiency or rather the inefficiency of the cooling devices of the compressor and the receiver, and not to the efficiency of the pipe line.

The maximum energy for doing useful work in expanding isothermally down to atmospheric pressure, which resides in a given weight of compressed air occupying a volume V_2, is the same as the energy expended in compressing isothermally that same weight of free air to a given pressure and delivering it under that pressure via the receiver into the pipe line.

Let V_1 = volume in cubic feet of a given weight of free air to be compressed, being at outside temperature.
P_1 = initial absolute pressure in pounds per square inch.
P_2 = final absolute pressure in pounds per square inch.
V_2 = volume in cubic feet of the same weight of air after being compressed isothermally to a pressure P_2.

Then the energy residing in this volume V_2 of compressed air after leaving the compressor and at the entrance of the pipe line is:

$$\text{Energy at entrance of pipe line} = 144\, P_1 V_1 \log_e \frac{V_1}{V_2}$$
$$= 144\, P_1 V_1 \log_e \frac{P_2}{P_1} \text{ foot-pounds.}$$

At the end of the pipe line this volume V_2 owing to the loss of pressure due to friction and other causes, has expanded into a volume V_3, whereas the pressure has decreased to a pressure P_3.

The energy residing in this volume V_3 at a pressure P_3 for doing useful work is the same as the energy that would have been expended in compressing isothermally a volume V_1 of free air from a pressure P_1 to a pressure P_3 and delivering the compressed air which now occupies a volume V_3 via the receiver into the pipe line.

That is, energy residing in the compressed air at the end of the pipe line:

$$W_n = 144\, P_1 V_1 \log_e \frac{V_1}{V_3} = 144\, P_1 V_1 \log_e \frac{P_3}{P_1} \text{ foot-pounds.}$$

Hence efficiency of pipe line

$$E = \frac{144\,P_1V_1\log_e\dfrac{P_3}{P_1}}{144\,P_1V_1\log_e\dfrac{P_2}{P_1}}$$

and since the modulus cancels

$$E = \frac{\log\dfrac{P_3}{P_1}}{\log\dfrac{P_2}{P_1}} \qquad (1)$$

As P_3 is always smaller than P_2 it would appear from formula (1) that the efficiency of a pipe line of certain dimensions becomes smaller, the higher the pressure P_2 is at which the compressed air enters the pipe line. This would be true if P_3 remained the same when P_2 is being increased.

According to equation (6), Art. 96, however, the terminal pressure and therefore the potential energy of the air at the discharge end of the pipe line increases with the increase of the initial pressure, and so does the pipe line efficiency.

The subsequent loss of energy due to the fact that the expansion of the compressed air in the air engine will be adiabatic instead of isothermal, must be charged to the efficiency of the air engine or to the whole system, but not to the pipe line.

Example.—What is the efficiency of a 6-in. pipe line, 1200 ft. long, delivering at its terminal a free-air equivalent of 4000 cu. ft. per minute at a pressure of 92 lb. gage? Atmospheric pressure = 14.7 lb.

Introducing in equation (5), Art. 96, the proper subscripts, the required initial pressure will be

$$P_2 = \sqrt{\frac{4000^2\times 1200}{2000\quad\times 6^5}+106.7^2} = 112.34 \text{ lb. abs.}$$

Then $\qquad \dfrac{P_3}{P_1} = \dfrac{106.7}{14.7} = 7.26$

and $\qquad \dfrac{P_2}{P_1} = \dfrac{112.34}{14.7} = 7.64$

Whence $\qquad E = \dfrac{\log 7.26}{\log 7.64} = 0.975$ or 98 per cent.

The pressure loss in this case is $112.34 - 106.70 = 5.64$ lb.

If the same quantity of air were transmitted through the same pipe

line with an initial pressure of 130 lb. abs. the end pressure according to equation (6), Art. 96, would be:

$$P_3=\sqrt{130^2-\frac{4000^2\times 1200}{2000\times 6^5}}=125.5 \text{ lb. abs.}$$

showing a pressure loss of only $130.0-125.5=4.50$ lb.

This indicates that, other things remaining the same, higher initial pressures result in higher pipe line efficiency.

101A. Résumé of Pipe Line Computations.—As has been pointed out, one of the chief difficulties of accurate pipe calculation lies in the fact that compressed air, while flowing through a pipe, creates friction which gradually but persistently decreases the pressure of the air. This loss of pressure is accompanied by a corresponding expansion of the air, that is, by an increase in volume. To transmit this increased volume through the pipe, requires increased velocity of flow resulting in a further increase of friction loss and so on.

Mathematical formulas for general use must of necessity assume more or less uniform conditions. In the pipe line formulas it is assumed that friction losses are directly proportional to the length of the pipe. That is, if a loss of 8 lb. occurs, for instance, in a pipe 5.85 in. in diameter and 1200 ft. long, transmitting a free-air equivalent of 4000 cu. ft. per min. (Example 1, Art. 96), then a loss of 16 lb. would be found from the formulas for a 2400 ft. pipe line of the same diameter and transmitting the same amount of air at the same initial pressure.

The assumption implies that the velocity of flow is the same at all points in the line, which would only be the case if the pipe line were constructed with a gradually increasing cross-sectional area, corresponding to the increase in volume mentioned above. This is, of course, impracticable in commercial operations.

Although the pressure loss due to friction increases with the length of the pipe, the number of elbows and other obstructions, the actual loss in any particular section of the pipe line is greater than that in the preceding one, not in direct proportion to the length of the pipe but in a proportion that eludes statement by simple mathematical formulas. The shorter the pipe line, the more nearly correct will be the results obtained from the formulas, because the more nearly uniform will be the conditions between the intake and the discharge terminal of the pipe line. The longer the pipe line, the greater will be the discrepancies between the results obtained by the formulas and those obtained in practice. Hence the necessity of making sufficient allowance for the many incalculable factors such as, for instance, the roughness of the pipe and the obstructions the air meets at each joint of the pipe line. Unsatisfactory results in long transmission lines are frequently due to failure to realize the magnitude of the pressure losses caused by these factors.

No general rule can be laid down as to the proper allowances to be made in each individual case. All that may be said is, that it usually pays to increase the diameter of the pipe beyond that obtained by calculation up to a point at which the economic advantages gained no longer balance the extra cost of installation.

102. Effect of Altitude on Pipe-line Efficiency.—In making estimates of pipe-line efficiency it must be borne in mind that so-called losses, due to difference of elevation, as explained under Article 97 cannot be charged to the efficiency of the pipe line. For the pressure loss, due to the difference of elevation is a constant quantity, no matter what the dimensions of the pipe line may be. This loss must be charged to the compressed-air installation as a whole as pointed out in Article 121.

103. Final Dimensions of Pipe Line.—In any individual installation, the length of the pipe line is generally a given quantity as well as the amount of air that must be delivered at a certain pressure at the end of it. We have seen that high initial pressures give high pipe-line efficiency, but require more powerful and more expensive compressors. On the other hand, with low initial pressure, a more expensive pipe line of larger diameter is required in order that the pressure at the discharge end may be a definite quantity.

Since the power residing in compressed air depends on the pressure as well as on the volume, and since in a pipe line the decrease in pressure is, up to a certain point, greater than the loss of power, it is a question whether it is more economical to have a high initial pressure and a smaller pipe line or a lower initial pressure and a larger pipe line.

In general, the decision in favor of the one or the other must be made by taking into consideration a number of factors and by comparing the costs for each individual case.

In making these computations it must also be borne in mind that the power required to compress air to a certain pressure is not in direct proportions to these pressures themselves as pointed out under Article 49.

Under certain conditions it may be more economical in the end to have high initial pressures with a smaller size pipe line. Under other conditions the reverse may be the case. This is a problem which can be only solved after due consideration of all the conditions which affect the plant to be installed, such as cost of pipes, cost of fuel, transportation, labor, etc., and the probable life of the plant.

104. Pipe-line Construction.—From what has been said regarding losses in the transmission of compressed air through pipes, it is evident that not only the design of a pipe line should

be given due attention, but that the laying of the line should also receive considerable care.

Friction being the chief cause of loss in an otherwise well-proportioned and constructed pipe line, it is desirable that the interior of the pipe should be as smooth as possible. In ordering pipes, particular mention should be made that the interior of the pipes should be free from all roughness such as scale, blisters, lumps, etc., and when the piping is put up, great care should be taken to clean the lengths thoroughly of dirt which may have gotten into them.

Where the line is exposed to severe cold, the moisture in the air will condense and the water so formed will freeze in the pipes until it throttles or chokes the pipe altogether. Since this takes place particularly in low points of the pipe line which form pockets for the accumulation of the entrained water, such pockets should be avoided as much as possible.

Valves and bends will increase the friction to à great extent. A globe valve causes the greatest loss and an ell or tee causes a loss of one-half to two-thirds that of a globe valve. Consequently care should be taken that gate valves be used instead of globe valves, and as few bends put in as possible. Where turns are absolutely necessary they should be made with as long a sweep as possible, either by bending the pipe without kinking it or by using long-sweep ells or tees.

Long pipe lines which are exposed to high temperatures should be provided with expansion joints to avoid springing leaks. Leaks should be attended to as soon as discovered. They cause the air in the pipes to expand and to lose pressure rapidly.

The heavy losses, caused by leaks in a pipe line will become clear by a study of Article 105 and the numerical example contained therein.

FLOW OF COMPRESSED AIR FROM AN ORIFICE INTO THE ATMOSPHERE

105. Let the confined air be under a pressure of "p" pounds gage. The theoretical velocity with which it flows from an orifice into the atmosphere is:

$$v = \sqrt{2gh} \text{ feet per second.} \tag{1}$$

in which v = velocity in feet per second.

g = acceleration due to gravity = 32.2 ft. per second.

h = height in feet of a column of air of uniform density, corresponding to a gage pressure p and exerting a pressure of p pounds, on its base which is assumed to be 1 sq. in. in area.

The volume V of this column is:

$$V=\frac{1}{144}\,h \text{ cubic feet.} \tag{2}$$

It is evident that the mass of air forming this column must weigh p pounds in order to exert a pressure of p pounds per square inch which is the area of its base.

According to Article 2 the weight W of 1 cu. ft. of atmospheric air at 60° Fahr. is 0.0764 lb. The weight W_1 of 1 cu. ft. of air having a density corresponding to a gage pressure p, we find from Article 22:

$$\frac{W_1}{W}=\frac{p+14.7}{14.7}$$

whence

$$W_1=0.0764\,\frac{p+14.7}{14.7}\text{ lb. per cubic foot.}$$

Since the total weight of the air column must be p pounds, its volume must be:

$$V=\frac{p}{0.0764\frac{p+14.7}{14.7}}\text{ cubic feet.} \tag{3}$$

Combining equations (2) and (3), we have:

$$\frac{h}{144}=\frac{p}{0.0764\frac{p+14.7}{14.7}}$$

whence

$$h=\frac{144p\times 14.7}{0.0764(p+14.7)}$$

or

$$h=27{,}707\frac{p}{p+14.7} \tag{4}$$

Introducing this value in equation (1) we get:

$$v=\sqrt{2\times 32.2\times 27{,}707\frac{p}{p+14.7}}$$

or, theoretical velocity $v = 1336\sqrt{\dfrac{p}{p+14.7}}$ feet per second (5)

The actual velocity is, of course, less owing to friction and other causes. It is obtained by multiplying the theoretical velocity by an orifice coefficient "c." For ordinary compressed air problems such as leaks in receivers and pipe lines, where the pressures range from five to ten atmospheres, this coefficient may be taken as

$$c = 0.50$$

This gives for actual velocity:

$$v = 1336 \times 0.50\sqrt{\frac{p}{p+14.7}}$$

or $$= 668\sqrt{\frac{p}{p+14.7}} \text{ feet per second.} \qquad (6)$$

The volume V_1 of compressed air in cubic feet per minute, that flows from an orifice of the area "a" square feet is:

$$V_1 = 60 \times a \times v = 60 \times 668 \times a\sqrt{\frac{p}{p+14.7}}$$

or $V_1 = 40{,}080 \times a\sqrt{\dfrac{p}{p+14.7}}$ cu. ft. of compressed air per min. (7)

After expansion to atmospheric or free air, and after having assumed outside temperature, the volume V_1 of compressed air will occupy a volume V_a which we find from Article 15 as follows:

$$\frac{V_a}{V_1} = \frac{P_1}{P_a}$$

whence $$V_a = V_1\frac{P_1}{P_a} \qquad (8)$$

in which P_a = atmospheric pressure in pounds per square inch = 14.7.

P_1 = absolute pressure of compressed air in pounds per square inch = $p+14.7$.

Introducing values in equation (8) we get:

$$V_a = \frac{40{,}080(p+14.7)}{14.7}\, a\sqrt{\frac{p}{p+14.7}}$$

or $$V_a = 2727\, a\sqrt{p(p+14.7)} \text{ cubic feet} \qquad (9)$$

of free air per minute at outside temperature.

in which a = area of orifice in square feet.
p = gage pressure of compressed air in pounds per square inch.

Example.—Air under pressure of 80 lb. gage escapes from various leaks in a pipe line. What is the quantity of escaping air, expressed in cubic feet of free air per minute, when the combined area of the leaks is 1/2 sq. in.?

In this case

$$a=\frac{0.50}{144}\text{ sq. ft.}$$

$$\text{Therefore } V_a=\frac{2727\times0.50}{144}\sqrt{80(80+14.7)}=824\text{ cu. ft.}$$

of free air per minute, having outside or in-take temperature.

If the compressor is built to furnish 1500 cu. ft. of free air per minute, the leaks in the pipe line will cut down its useful capacity to less than one-half, without, however, cutting down the power required to run it.

The theoretical horse-power required to compress in two stages 824 cu. ft. of free air per minute to 80 lb. gage and deliver it into the receiver, is (from column 7, Table V)

$$824\times0.141=116\text{ h.p.}$$

which represents the theoretical power loss, due to the leaks, which at first sight seem rather insignificant. This points to the importance of stopping them as soon as discovered.

PART III

THE USE OF COMPRESSED AIR

CHAPTER XII

THEORY OF AIR ENGINES

106. Compressed air can be used to operate an engine in a manner similar to steam, either at full pressure or expansively. In the first case air at full pressure is admitted into the cylinder of the engine during the entire stroke and is exhausted practically at full pressure. In the second case air is admitted into the cylinder during part of the stroke, is then cut off and used expansively for the remainder of the stroke. In practice it is always exhausted at a pressure slightly above atmospheric pressure for reasons stated in Article 112.

107. Compressed Air Used at Full Pressure During the Entire Stroke.—Although the efficiency of an engine using air in this fashion is, of necessity, small, the waste of energy in such engines is usually compensated in part by the saving in first cost and labor, due to the simplicity of construction of such machines and the ease of handling them.

108. The theoretical net work in foot-pounds performed per stroke by engines using air at full pressure during the entire stroke is equal to the total force multiplied by the distance through which it acts.

The total force is: (absolute pressure of compressed air on the in-take side minus atmospheric pressure on the exhaust side) multiplied by (area of piston). The distance through which the force acts is the length of the stroke.

Let W_n = net work in foot-pounds.
P_1 = absolute pressure of air in pounds per square inch on in-take side.
P_a = atmospheric pressure in pounds per square inch on exhaust side.
A = area of piston in square feet.
L = length of stroke in feet.
V_1 = volume of compressed air in cubic feet taken into the cylinder per stroke.

Then $\qquad W_n = 144\ (P_1 - P_a) \times AL$

And since $\qquad AL = V_1$

$$W_n = 144\ (P_1 - P_a) V_1 \text{ foot-pounds.}$$

Assume that the volume V_1 of air required to do this work is obtained by isothermal compression, then the work of supplying this volume would be a minimum, viz:

$$144\ P_a V_a \log_e \frac{P_1}{P_a} \text{ foot-pounds}$$

in which $V_a =$ volume of free air which, after being compressed isothermally to an absolute pressure P_1 would occupy a volume V_1

Therefore $\qquad V_a = V_1 \frac{P_1}{P_a}$

109. Maximum efficiency of air engine using air at full pressure during the entire stroke is:

$$E = \frac{144\ (P_1 - P_a)\ V_1}{144 P_a V_1 \frac{P_1}{P_a} \log_e \frac{P_1}{P_a}} = \frac{P_1 - P_a}{P_1 \log_e \frac{P_1}{P_a}}$$

Dividing dividend and divisor by P_a

$$E = \frac{\frac{P_1}{P_a} - 1}{\frac{P_1}{P_a} \log_e \frac{P_1}{P_a}}.$$

This shows that the higher the initial pressure P_1 at which the air enters the air engine, the smaller becomes the efficiency.

Example.—A striking example of an apparatus using air at full pressure during the entire stroke is the well-known rock drill. In consumption of power it is one of the most wasteful machines, but from a practical point of view its efficiency stands at present unquestioned.

The theoretical efficiency of a rock drill with a cylinder of 3 1/4 in. in diameter, 6 3/4-in. stroke, 400 strokes per minute, using air at 60 lb. gage is as follows:

Area of piston.............................. 8.29 sq. in.

Piston displacement $8.29 \times 6\ 3/4 =$ .... 55.96 cu. in.

Volume of air taken into cylinder per minute

$\frac{55.96}{1728} \times 400 =$ .... 12.95 cu. ft. at 60 lb.

Volume of free air, equivalent to 12.95 cu ft. at 60 lb.

$V_a = 12.95\ \frac{60 + 14.7}{14.7} =$ .... 65.8 cu. ft. per min.

Theoretical horse-power required to compress and deliver 65.8 cu. ft. per min. at 60 lb. gage (from column 4, Table V) $65.8 \times 0.134 =$ ..... 8.8 h.p.
Power utilized in striking rock (forward stroke only), $8.29 \times 60 \times \frac{6\ 3/4 \times 200}{12} =$ .... 56,200 ft.-lb.
$=$ 1.70 h.p.

In this case the theoretical efficiency of the drill is only 19 per cent.

The practical efficiency will be much less, due to friction leakage, etc., in the drill itself and to power losses in bringing the air to the drill.

COMPRESSED AIR USED WITH COMPLETE ADIABATIC EXPANSION

110. The theoretical net work performed per stroke by engines using air with complete adiabatic expansion down to atmospheric pressure, is deduced in the same manner as that for compression.

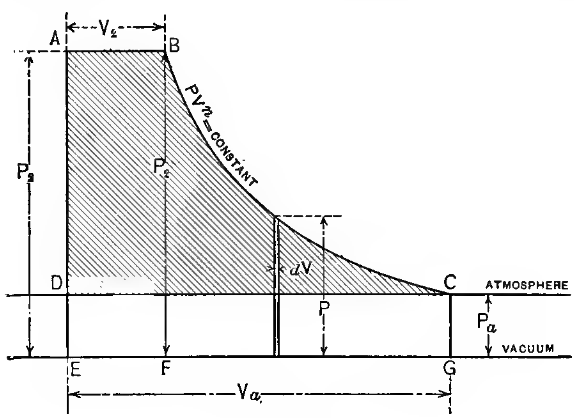


FIG. 15.

Referring to Fig. 15:

$$W_n = 144 \times (\text{shaded area } ABCD) \text{ foot-pounds}$$

But area $ABCD$ = area $ABFE$
plus area $FBCG$
minus area $EDCG$

$$\text{Area } ABFE = P_2V_2$$

$$\text{Area } FBCG = \int_{V_2}^{V_a} P\, dV$$

But $$PV^n = P_2V_2^n \text{ or } P = \frac{P_2V_2^n}{V^n}$$

hence area $FBCG$ $$= \int_{V_2}^{V_a} \frac{P_2V_2^n}{V^n}\, dV$$

$$= P_2V_2^n \int_{V_2}^{V_a} \frac{dV}{V^n}$$

$$= P_2\,V_2^n \int_{V_2}^{V_a} V^{-n} dV$$

$$= P_2V_2^n \frac{V_a^{1-n} - V_2^{1-n}}{1-n}$$

$$= \frac{P_2V_2^nV_a^{1-n} - P_2V_2^nV_2^{1-n}}{1-n}$$

and since $P_2V_2^n = P_aV_a^n$
we can write:

area $FBCG$ $$= \frac{P_aV_a^nV_a^{1-n} - P_2V_2^nV_2^{1-n}}{1-n}$$

$$= \frac{P_aV_a - P_2V_2}{1-n}$$

$$= \frac{P_2V_2 - P_aV_a}{n-1}$$

area $EDCG$ $$= P_aV_a$$

therefore $$W_n = 144\ \left(P_2V_2 + \frac{P_2V_2 - P_aV_a}{n-1} - P_aV_a\right)$$

$$= \frac{144\ n}{n-1} P_2V_2 \left[1 - \frac{P_aV_a}{P_2V_2}\right]$$

and since $$\frac{V_a}{V_2} = \left(\frac{P_2}{P_a}\right)^{\frac{1}{n}}$$

$$W_n = \frac{144\ n}{n-1} P_2V_2 \left[1 - \left(\frac{P_a}{P_2}\right)^{\frac{n-1}{n}}\right] \text{ foot-pounds.} \quad (1)$$

in which W_n = net work in foot pounds per stroke.
P_2 = absolute pressure in pounds per square inch of compressed air entering cylinder.
V_2 = volume of compressed air in cubic feet taken into the cylinder per stroke.
P_a = atmospheric pressure in pounds per square inch.
n = 1.406.

111. The theoretical horse-power which a volume V_2 of air in cubic feet per minute, compressed to an absolute pressure P_2 is capable of developing during admission and adiabatic expansion to atmospheric pressure, is obtained by letting V_2 in the formula for W_n represent the number of cubic feet per minute admitted into the cylinder and by dividing the whole by 33,000.

$$\text{Theoretical horse-power} = \frac{144nP_2V_2}{33{,}000\ (n-1)}\left[1-\left(\frac{P_a}{P_2}\right)^{\frac{n-1}{n}}\right] \qquad (1)$$

in which P_2 = absolute pressure in pounds per square inch of air entering cylinder.
V_2 = volume of compressed air taken into the cylinder in cubic feet per minute.
P_a = atmospheric pressure in pounds per square inch.
n = 1.406.

112. In practice complete expansion to atmospheric pressure is not feasible for the following reasons:

1. The resulting increase in volume of the expanded air would require a more expensive engine with larger cylinders than is warranted by the small gain in power.

2. To overcome the friction of the engine near the end of the stroke and to cause the air to properly exhaust against the back pressure of the atmosphere, the pressure of the exhaust air must be somewhat greater than the atmospheric pressure.

3. Unless the compressed air is reheated before being used, its temperature when entering the air cylinder of the engine is that of the surrounding atmosphere. A high ratio of expansion will result in very low final temperature and quite often in freezing of the moisture in the air around the exhaust ports.

Example.—Assuming that air at 80 lb. gage and 60° Fahr. enters the cylinder of an air engine and that it is allowed to expand adiabatically

to atmospheric pressure at sea level, then the theoretical final absolute temperature of the exhaust air would be:

$$T_1 = T\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} = (60+461)\left(\frac{14.7}{94.7}\right)^{0.29} = 305° \text{ absolute.}$$
$$= -156° \text{ Fahr.}$$

COMPRESSED AIR USED WITH PARTIAL ADIABATIC EXPANSION

113. For reasons stated in Article 112, air engines usually work with partial expansion, that is, a volume of compressed air is admitted during part of the stroke, is then cut off and allowed to expand down to a pressure somewhat above atmospheric pressure.

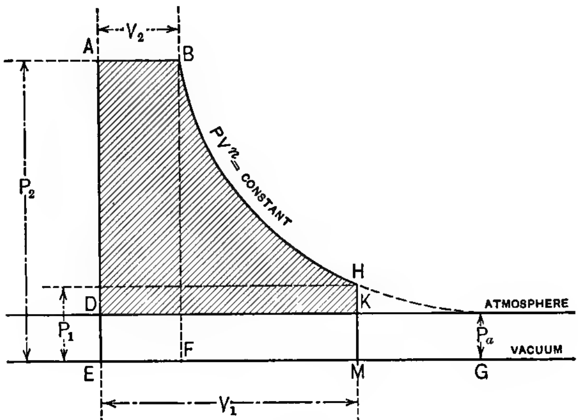


FIG. 16.

Referring to Fig. 16 the net work performed during one stroke of the piston is:

$$W_n = 144 \times (\text{shaded area } ABHKD)$$

But area $ABHKD$ = area $EABF = P_2V_2$

plus area $FBHM = \dfrac{P_2V_2 - P_1V_1}{n-1}$

minus area $DEMK = P_aV_1$

whence $W_n = 144\left(P_2V_2 + \dfrac{P_2V_2 - P_1V_1}{n-1} - P_aV_1\right)$ foot-pounds, (1)

in which P_2 = initial absolute pressure in pounds per square inch of air entering engine.

V_2 = volume of compressed air in cubic feet taken into the cylinder per stroke.
P_1 = absolute pressure in pounds per square inch of exhaust air.
V_1 = volume of air exhausted per stroke at a pressure P_1.
P_a = atmospheric pressure in pounds per square inch.
n = 1.406.

114. Mean gage pressure of air during admission and partial adiabatic expansion:

$$P_m = \frac{W_n}{144\ V_1} = \left[P_2\frac{V_2}{V_1} + \frac{P_2\frac{V_2}{V_1} - P_1}{n-1} - P_a \right] \text{ pounds per square inch.}$$

115. Theoretical horse-power which a volume of compressed air per minute is capable of developing during admission and partial adiabatic expansion:

$$\text{Horse-power} = \frac{144}{33{,}000}\left[P_2V_2 + \frac{P_2V_2 - P_1V_1}{n-1} - P_aV_1\right]$$

in which, P_2 = absolute pressure in pounds per square inch of in-take air.
V_2 = volume of compressed air in cubic feet taken into the cylinder per minute.
P_1 = absolute pressure in pounds per square inch of exhaust air.
V_1 = volume of air exhausted at a pressure P_1 in cubic feet per minute.
P_a = atmospheric pressure in pounds per square inch.
n = 1.406.

In practical problems P_2, V_2 and P_a are always known, and either P_1 or V_1 are given.

If P_1 is given, then V_1 is found from the relation:

$$\frac{V_1}{V_2} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

whence
$$V_1 = V_2\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

If V_1 is given, as in engines having a certain cut-off, then P_1 is found from the relation:

$$\frac{P_1}{P_2}=\left(\frac{V_2}{V_1}\right)^n$$

whence
$$P_1=P_2\left(\frac{V_2}{V_1}\right)^n$$

Example.—Find theoretical horse-power developed by 1 cu. ft. of air per minute, having a pressure of 100 lb. gage, being admitted to and expanded adiabatically in an air engine with 1/4 cut-off. Atmospheric pressure = 14.7 lbs.

For 1/4 cut-off $V_1=4V_2=4$ cu. ft.

and $P_1=(100+14.7)(1/4)^{1.406}=16.3$ lb. per sq. in.

$$\text{Horse-power}=\frac{144}{33{,}000}\left[114.7+\frac{114.7-16.3\times4}{0.406}-14.7\times4\right]=0.733.$$

116. Modified Power Values for Practical Air-engine Problems.—In practical computations, the theoretical formulas, expressing the energy residing in a quantity of compressed air for doing useful work, must be modified for the same reasons explained in Article 74.

It is customary to subtract 15 per cent. from the theoretical values in order to obtain the actual work that an air engine may be expected to perform under normal conditions.

CHAPTER XIII

EFFECT OF LOSS OF HEAT, GENERATED DURING COMPRESSION, ON THE ULTIMATE USEFUL ENERGY RESIDING IN A GIVEN QUANTITY OF COMPRESSED AIR

117. By an accepted law of thermodynamics, work and heat are mutually convertible at the ratio of about 778 ft.-lb. of work for every B.T.U.

In Article 41a it was stated that the work expended in compressing air is all converted into heat. According to the law quoted, we should expect the compressed, and therefore heated, air to be capable of performing useful work, equal to the amount expended in compressing it. Neglecting friction in the air engine, this would actually be the case, if the compressed air could be used immediately after compression and before it has lost any of its heat.

If, on the other hand, the compressed air be allowed to cool down to the temperature which it possessed before compression, as happens in all compressed air installations, it would seem logical, by applying the same law quoted above, to reason as follows:

Since the work of compression is all converted into heat, the ability for doing useful work must have disappeared after all this heat has been abstracted.

In the following articles it will be shown:

a. That the work of compression is all converted into heat.

b. That, after all the heat of compression has been abstracted, there still remains in the compressed air a certain amount of energy for doing useful work.

c. That this is due to the energy residing in the air before compression.

a. Referring to Fig. 17, the total work of compressing adiabatically a volume V_1 cubic feet of free air from an absolute pressure P_1 to an absolute pressure P_2 is represented by the area $MABR$. Expressed in foot-pounds, it is equal to 144 times the numerical value of this area.

In Article 44 we found:

$$\text{Area } MABR = \frac{P_2V_2 - P_1V_1}{n-1} \tag{1}$$

therefore, total work of compression

$$W_1 = 144\frac{P_2V_2 - P_1V_1}{n-1} \text{ foot-pounds.} \tag{2}$$

Let P_1 = 14.7 lb. absolute pressure per square inch.
P_2 = 89.7 lb. absolute pressure per square inch.
= 75 lb. gage.
V_1 = 13.09 cu. ft. which is the volume of 1 lb. of free air at sea level and at 60° Fahr.
n = 1.406.

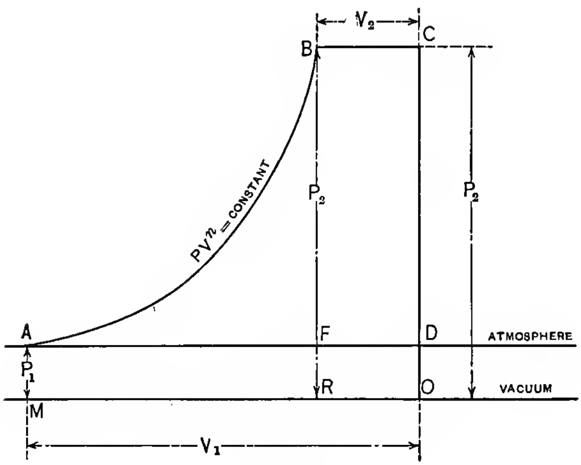


FIG. 17.

From equation (7), Article 41, deduce:

$$\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

whence $$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} = 13.09\left(\frac{14.7}{89.7}\right)^{0.71} = 3.62 \text{ cu. ft.} \tag{3}$$

Substituting values in equation (2) we get:

$$W_1 = 144\frac{89.7 \times 3.62 - 14.7 \times 13.09}{0.406} = 47{,}000 \text{ ft.-lb.} \tag{4}$$

After the air has been compressed adiabatically to an absolute pressure P_2 its absolute temperature will be according to equation (11), Article 41:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = (60+461)\left(\frac{89.7}{14.7}\right)^{0.29} = 880° \text{ absolute} \qquad (5)$$
$$= 419° \text{ Fahr.}$$

After compression, the original pound of air occupies a volume $V_2 = 3.62$ cu. ft. and has a temperature of 419° Fahr. which is $(419-60) = 359$ degrees more than its initial temperature.

Now, we can imagine a volume V_2 of air weighing 1 lb. to have a temperature of 60° Fahr. If we raise the temperature of this air by $(T_2 - T_1) = (880-561) = 359$ degrees without changing its volume, we heat under constant volume. The specific heat C_v of air in this case is 0.168 and the amount of heat put into this pound of air, expressed in B.T.U.'s. is

$$C_v(T_2 - T_1) = 0.168 \times 359 = 60.3 \text{ B.T.U.'s.}$$

Expressed in foot-pounds it is:

$$K_v(T_2 - T_1) = 131.6 \times 359 = 47{,}000 \text{ ft.-lb.} \qquad (6)$$

A comparison of equation (6) with (4) shows that the mechanical equivalent of the heat required to raise the temperature of 1 lb. of air from an absolute temperature T_1 to an absolute temperature T_2 is identical with the mechanical energy expended in compressing adiabatically 1 lb. of atmospheric air having an absolute temperature T_1 to a pressure which raises the temperature of the air to an absolute temperature T_2. In other words, the mechanical work of compressing air adiabatically is all converted into heat energy.

b. If we now allow this volume $V_2 = 3.62$ cu. ft. of compressed air, having a temperature of 419° Fahr., to cool down to initial temperature of 60° Fahr. under constant volume, its pressure will decrease to a pressure P_3, which we find from the formula:

$$P_3 = P_2\frac{T_3}{T_2} = 89.7 \times \frac{521}{880} = 53.2 \text{ lb. absolute.}$$

The energy residing in this volume $V_3 = 3.62$ cu. ft. of air for doing useful work in expanding adiabatically down from an absolute pressure of 53.2 lb. to atmospheric pressure is represented

by the area *BCGF* in the diagram, Fig. 18, and expressed in foot-pounds it is 144 times the numerical value of this area. From article 110 we deduce:

$$\text{Area } BCGF = \frac{P_3V_2 - P_1V_1}{n-1}$$

Hence energy $$W = 144\frac{P_3V_2 - P_1V_1}{n-1}$$

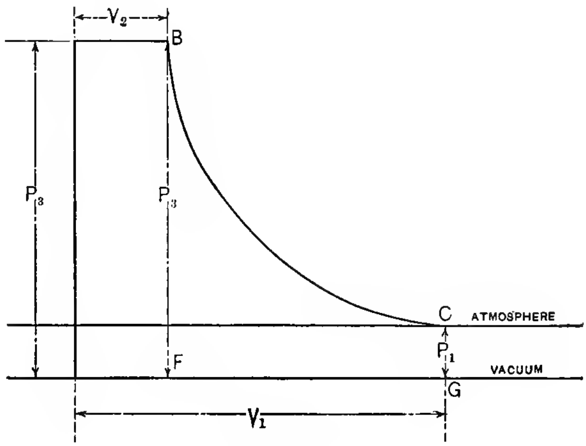


FIG. 18.

Applying it to the case in hand:

P_3 = 53.2 lb. absolute per sq. in.

V_2 = 3.62 cu. ft.

P_1 = 14.7 lb. per sq. in.

$V_1 = V_2\left(\frac{P_3}{P_1}\right)^{\frac{1}{n}} = 3.62\left(\frac{53.2}{14.7}\right)^{0.71} = 9.02$ cu. ft. (From equation 13, Article 41.)

n = 1.406.

Hence $W = 144 \times \frac{53.2 \times 3.62 - 14.7 \times 9.02}{0.406} = 21{,}300$ ft.-lb.

Comparing this with the work of compression, we have:

$$\frac{21{,}300}{47{,}000} = 0.45 = 45 \text{ per cent.}$$

That is, theoretically, after cooling down to initial temperature, there still remains in the compressed air energy for doing expansive work to the amount of 45 per cent. of the energy expended in compressing it.

Referring to the diagram in Fig. 17, it will be noted that part of the total work of compression represented by the area *MABR* is performed by the atmospheric air rushing into the cylinder behind the piston during the compression stroke and not by energy furnished by the compressor. This work is represented by the area *MAFR*.

In practice, the air, after being compressed, is delivered into the receiver. The work of delivery is jointly performed by the compressor and by the atmospheric air. The compressor's work is represented by the area *FBCD* and the work of the atmosphere by the area *RFDO*. The net work of compression and delivery done by the air compressor alone is represented by the area *ABCD*. The compressor's share of delivery work is always available for doing useful work in the air engine because in forcing a volume of compressed air from the air-cylinder into the receiver, an equal volume of air is displaced therein, and this displacement process is extended into the pipe line and finally into the air engine, where, in making room for itself, this volume of compressed air drives the piston forward, and thus does useful work.

It may be asked: What becomes of the energy contributed by the atmospheric air toward compression and delivery which is represented by the area *MADO* in Fig. 17?

This energy is actually stored up in the compressed air when the latter leaves the compressor. It could do useful work if it were practicable to exhaust the air from the engines into a vacuum. But since we must exhaust against atmospheric pressure, the energy is consumed in the process of exhaustion and is therefore not available for useful work. It is not included in the formulas expressing power to be furnished by the compressor because it is furnished gratis by the atmosphere; and it is not included in the formulas expressing the useful work which a volume of compressed air can perform, because it is not available for such work.

The following example shows the effect of heat loss upon the total power stored up in a mass of air by the compressor.

Example.—To compress adiabatically in one stage 100 cu. ft. of free air per minute at sea level to 60 lb. gage and deliver it into the receiver, requires (theoretically) 13.40 h.p. (from column 4 Table V).

If the temperature of the free air was 60° before compression, after compression it will be 375° Fahr. (column 6 Table V) and the volume of the compressed air will be 31.44 cu. ft. (column 5 Table III)

If used immediately after compression, before having lost any heat, it could do work (theoretically) to the amount of 13.40 h.p. by expanding adiabatically down to atmospheric pressure.

But if allowed to cool, before use, to initial temperature under constant volume, the pressure will decrease to a pressure P_3 which we find from the following formula:

$$P_3 = P_2 \frac{T_3}{T_2} = (60+14.7)\frac{60+461}{375+461} = 46.6 \text{ lb. absolute.}$$

A volume of 31.44 cu. ft. of air per minute at 46.6 lb. absolute, if allowed to expand adiabatically down to atmospheric pressure could perform (theoretically) an amount of work found from equation (1) Article 111:

$$\text{Horse-power} = \frac{144\, n\, P_2 V_2}{33{,}000\, (n-1)}\left[1-\left(\frac{P_a}{P_2}\right)^{\frac{n-1}{n}}\right]$$

$$= \frac{144\times 1.406\times 46.6\times 31.44}{33{,}000\times 0.406}\left[1-\left(\frac{14.7}{46.6}\right)^{0.29}\right] = 6.30 \text{ h.p.}$$

which is about 47 per cent. of the power expended in compression and delivery.

When friction and other imperfections are taken into account, this percentage decreases materially.

Adding 15 per cent. to the power of production we get 15.43 h.p.

Subtracting 15 per cent. from the available theoretical energy we get 5.35 h.p. and the comparative value shrinks to 35 per cent. This is further diminished by losses during transmission which are pointed out under Articles 93–94 and 97–105.

c. The answer to the question, why energy still remains in the compressed air after all the heat of compression has been dissipated, is that a certain capacity for work resides in the air which is due to the latter's ability to expand when the proper conditions prevail.

Such conditions could be brought about by confining a volume of atmospheric air in a cylinder under a piston and then create a partial vacuum on the other side of the piston; the atmospheric air in the cylinder would expand and push out the piston, that is, perform work. But creating a vacuum requires extra work, and is therefore not of practical application in air engines.

As a matter of fact, after all the heat generated during compression of a volume of air has been dissipated, the compressed air possesses no more energy than it did before compression, but

part of the energy which it did possess has, by mechanical compression, been made available for doing useful work.

To do work, however, the air requires energy in the form of heat and while expanding, it consumes heat that was contained in its mass before compression. As a consequence the temperature of the expanded air falls below that of the surrounding atmosphere. The amount of heat consumed is equivalent to the amount of work performed and equal to the amount of heat that would be generated in compressing this air from the pressure at which it exhausts from the air engine to the pressure at which it enters the same.

The consumption of heat from the mass of the expanding air is manifested by the cold created in and around the cylinders of an engine using air expansively. Theoretically this is exactly the reverse of the generation of heat in the air cylinders of a compressor.

117a. Determination of the Value of "n," used in adiabatic compression and expansion formulas:

From equation (6), Article 117, we have:
Work of adiabatic compression of 1 lb. of free air:

$$W = K_v(T_2 - T_1) \text{ foot-pounds} \qquad (1)$$

in which K_v = specific heat of air at constant volume, expressed in foot-pounds.
T_2 = final absolute temperature of air after being compressed to an absolute pressure P_2.
T_1 = initial absolute temperature of air at an absolute pressure P_1.

In the diagram, Fig. 17, the area $MABR$ represents the mechanical work of compressing a volume V_1 of air from an absolute pressure P_1 to an absolute pressure P_2, the volume of compressed air being V_2.

From equation (1) Article 117:

$$\text{Area } MABR = \frac{P_2V_2 - P_1V_1}{n-1} \qquad (2)$$

Let P_1 and P_2 be the absolute pressures in pounds per square foot; then the work performed, corresponding to area $MABR$:

$$W = \frac{P_2V_2 - P_1V_1}{n-1} \text{ foot-pounds} \qquad (3)$$

Let, furthermore, V_1 and V_2 represent volumes occupied by 1 lb. of air when under an absolute pressure of P_1 or P_2 respectively; then from equation (5) Article 20:

$$P_1V_1 = RT_1$$

and

$$P_2V_2 = RT_2$$

Substituting these values in equation (3) we have:

$$W=\frac{RT_2-RT_1}{n-1}=\frac{R(T_2-T_1)}{n-1} \tag{4}$$

From equation (7) Article 20 we have:

$$R=K_p-K_v$$

Substituting in equation (4) we get:

$$W=\frac{(K_p-K_v)(T_2-T_1)}{n-1} \tag{5}$$

This work is equal to the work expressed by equation (1), therefore:

$$K_v(T_2-T_1)=\frac{(K_p-K_v)(T_2-T_1)}{n-1}$$

or

$$nK_v-K_v=K_p-K_v$$

whence

$$n=\frac{K_p}{K_v} \tag{6}$$

as first stated under Article 40.

CHAPTER XIV

INTERNAL OR INTRINSIC ENERGY OF AIR

118. A capacity for doing useful work by expanding against an external resistance, resides in a mass of air as long as its temperature is above the absolute zero. A pound of atmospheric air at 60° Fahr. at sea level, for instance, may be conceived as the outcome of a pound of air at the temperature of absolute zero to which a sufficient amount of heat has been supplied to raise its temperature by $(461+60)=521°$ Fahr., and its pressure to 14.7 lb. above the vacuum.

According to a law of thermodynamics, quoted in previous articles, the heat energy in this pound of air, corresponding to a temperature of 521° above the absolute zero, may be converted into mechanical energy whenever the conditions permit it. The capacity of air of performing work, due to its temperature above the absolute zero, is called *the internal or intrinsic energy of air*. It is independent of pressure, that is, a pound of atmospheric air at a temperature of 60° Fahr., has the same intrinsic energy as a pound of air under a pressure of 100 lb. having the same temperature of 60° Fahr. (See Articles 119 and 120.)

When applied to practice, there is a vast difference, however, between the pound of atmospheric air and the pound of air at 100 lb. pressure. In the first case none of the intrinsic energy residing in the air is available for useful work under ordinary conditions, whereas in the second case a portion of the intrinsic energy has by mechanical compression been made available for such work.

This may be better understood by comparison with the more familiar generation of water-power. Water flowing down a river possesses intrinsic energy, that is, a capacity for doing useful work when the proper conditions exist. These conditions are brought about by building a dam across the river which raises the water level and thus produces a head, the height of which, together with the amount of water delivered, determines the amount of useful work the water is capable of performing. By building the dam we have added nothing to the intrinsic

energy of the water, we have only made available a portion of that energy for performing useful work.

In an analogous manner, by compressing air isothermally, we add nothing to its intrinsic energy, we merely make a portion of that energy available for doing useful work. In actual practice, compression is more or less adiabatic, imparting heat energy to the air, which, however, is subsequently lost in transmission. The condition of the air before use is therefore the same as after isothermal compression.

The conception of internal or intrinsic energy indicates that when air expands without doing work, it loses none of its heat, because the intrinsic energy remains unchanged. The truth of this fact was first proved experimentally by Joule and the fact itself is known as Joule's Law.

119. Intrinsic Energy of a Pound of Atmospheric Air at a Temperature of 60° Fahr.—The specific heat of air under constant pressure is 0.2375, therefore the quantity of heat, that is, the number of B.T.U.'s required to raise the temperature of 1 lb. of atmospheric air from absolute zero to 60° Fahr. is:

$$(461+60)\times 0.2375=123.74 \text{ B.T.U.'s}$$

and the amount of work corresponding to this quantity of heat is $123.74\times 778=96{,}268$ ft.-lb. This is the intrinsic energy of 1 lb. of atmospheric air at 60° Fahr., none of which, however, is available for useful work under ordinary circumstances.

120. Intrinsic Energy of a Pound of Air at 100 lb. Gage and at 60° Fahr.—If permitted to expand adiabatically down to atmospheric pressure against an external resistance, this pound of air would perform work and therefore consume an amount of heat equal to the amount that was generated during adiabatic compression. The theoretical temperature of the air after expansion is deduced from formula (11) Article 41:

$$T_1=T\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}}=(60+461)\left(\frac{14.7}{100+14.7}\right)^{0.29}$$
$$= 286.55 \text{ degrees absolute.}$$
$$= -174.45° \text{ Fahr.}$$

The drop in temperature is therefore $(60+174.45)=234.45$ degrees and the number of B.T.U.'s consumed during expansion would be $234.45\times 0.2375=55.68$ B.T.U.'s.

The equivalent of 55.68 B.T.U.'s expressed in foot-pounds is $55.68 \times 778 = 43,321$ ft.-lb. This is the amount of intrinsic energy residing in the pound of compressed air which is available for doing useful work.

But there still remains energy in the air which might be used if it were possible for the air to expand down to the absolute zero of pressure, in which case the temperature of the air would drop from 286.55 absolute to the absolute zero of temperature. This represents a consumption of heat units equivalent to $(286.55 \times 0.2375) = 68.065$ B.T.U.'s and these 68.056 B.T.U.'s present work equivalent to $(68.056 \times 778) = 52,947$ ft.-lb. This latter energy is not available for useful work under ordinary circumstances.

The total intrinsic energy of the pound of air at 100 lb. gage and 60° Fahr. is $(43,321 + 52,947) = 96,268$ ft.-lb. which is the same as the total intrinsic energy of the pound of atmospheric air at 60° Fahr.

CHAPTER XV

THE EFFICIENCY OF A COMPRESSED-AIR SYSTEM

121. This is evidently the ratio between the ultimate work performed by the engine using compressed air and the power required to compress that air in the compressor.

In computing this efficiency all the possible losses must be taken into consideration which may occur from the moment a certain quantity of air enters the compressor until it is exhausted from the air engine.

These losses are chargeable:

1. To air being taken into the compressor from the engine room rather than from a cooler place. This results in a lesser quantity (weight) of air being taken into the cylinder per stroke, thereby increasing the power required to compress a given quantity of air per unit of time. This loss can be prevented by making adequate provisions for the air in-take from the coolest outside place around the compressor building (see Article 87).

2. To friction in the compressor. This will amount ordinarily to a power loss of from 15 to 20 per cent. It can be reduced by good workmanship to about 10 per cent. but cannot be avoided altogether.

3. To a series of imperfections in the compressing cylinders, such as insufficient supply of free air, difficult discharge, defective cooling arrangments, poor lubrication, etc.

4. To heat generated during compression which increases the power required for compressing a given quantity of air, for which there is no return, as the heat is afterward dissipated in transmission.

5. To loss of pressure in the pipe line, due to friction, etc.

6. To friction and fall of temperature during expansion of the air in the cylinder of the air engine.

7. To leaks in the compressor, the pipe line, and in the air engine.

Example.—Let us follow a volume of 10 cu. ft. of free air, having an initial temperature of 60° Fahr., from the moment it is taken into the cylinder of the compressor until it is exhausted from the air engine.

Referring to the example under Article 87, we found that it took 55,600 ft.-lb. of work to compress these 10 cu. ft. of free air in one stage to 70 lb. gage, and that we finally deliver into the pipe line a volume of 2.68 cu. ft. of air having an absolute pressure of 50 lb. and a temperature of 60° Fahr.

Assuming that the pipe line is so dimensioned that the loss of pressure is 5 lb., then the compressed air which is delivered at the end of the pipe line has an absolute pressure of 45 lb. and its volume has expanded at constant temperature to

$$V_2 = V_1 \frac{P_1}{P_2} = 2.68 \frac{50}{45} = 2.98 \text{ cu. ft.}$$

According to equation (1) in Article 113, a volume of 2.98 cu. ft. of air at 45 lb. absolute if admitted to and allowed to expand adiabatically down to say, 16.5 lb. absolute in an air engine, is capable of doing useful work (theoretically) to the amount of:

$$W_n = 144 \left[P_2V_2 + \frac{P_2V_2 - P_1V_1}{n-1} - P_aV_1 \right] \text{ foot-pounds}$$

in which $P_2 = 45$ lb. per square inch

$V_2 = 2.98$ cu. ft.

$P_1 = 16.5$ lb. per square inch

$V_1 = V_2 \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} = 6.08$ cu. ft.

$P_a = 14.7$ lb. per square inch

$n = 1.406$

$$W_n = 144 \left[45 \times 2.98 + \frac{45 \times 2.98 - 16.5 \times 6.08}{0.406} - 14.7 \times 6.08 \right]$$
$$= 18{,}400 \text{ ft.-lbs.}$$

Deducting 15 per cent. for friction, etc., gives:

Work performed by air engine, 15,600 ft.-lb.

Work of compression and delivery, 56,600 ft.-lb.

$$\text{Efficiency of whole system} = \frac{15{,}600}{56{,}600} = 28 \text{ per cent.}$$

Practical tests frequently show lower efficiencies than those obtained by calculation, due to leaks in compressor, pipe line, air engine, and to other imperfections. If air is used in the motor at full pressure during the entire stroke as, for instance, in air drills (see Article 109), the efficiency sinks to its lowest level.

As has been pointed out, the use of compound compressors with adequate cooling devices and the use of higher initial pressures will result in higher mechanical efficiency of the whole system. One of the means employed at present to increase this efficiency is a system known as "Reheating" which is described under Article 122.

CHAPTER XVI

REHEATING OF COMPRESSED AIR

122. In preceding articles it has been shown that the available energy residing in a given weight of compressed air at the end of the pipe line is considerably less than that which the same weight of compressed air could develop immediately after leaving the compressor. This is due to the fact that the volume of a given weight of air, having a given pressure, is smaller at the lower temperature which prevails at the end of the pipe line than that which it occupies when leaving the compressor at a high temperature, and to the fact that the available power residing in compressed air is dependent on volume as well as on pressure.

This has led to the introduction of a process known as "Reheating." By this process the volume of the compressed air at the terminal may be, by heating, increased so as to partly or completely make up for loss of power in transmission.

To accomplish this result there must be an expenditure of fuel. This expense, however, is very light. For the average air engine it amounts to about one-seventh of the fuel that would be originally required to compress air so that it would be in a condition to develop an equal power, it being assumed that coal is the fuel used. This is due to the fact that the average efficiency of a Corliss steam engine does not exceed 10 per cent., based on the total heat value of the fuel, whereas in reheating in a proper heater 70 per cent. of the heat value of the fuel may be utilized. The increase in efficiency resulting from reheating makes it possible to use a much smaller air compressor for performing a given amount of work.

In addition to increasing the efficiency, the reheating of compressed air also prevents the freezing of the exhaust ports of air engines which often becomes troublesome when air containing considerable moisture is exhausted at temperatures below the freezing point.

Let us assume that at the end of the pipe line we have 1 cu. ft. of air at 75 lb. gage, and at a temperature of 60° Fahr. and that we wish to double this volume by reheating.

The volume of free air which must be compressed to make, after being cooled down to 60° Fahr., 1 cu. ft. of air at 75 lb. gage, we find from the equation

$$V=V_1\frac{P_1}{P}$$

whence $$V=1\times\frac{89.7}{14.7}=6.10 \text{ cu. ft.}$$

To compress adiabatically in one stage 6.10 cu. ft. of air to 75 lb. gage and deliver it into the receiver or the pipe line requires work to the amount of

$$W_n=\frac{144\,nPV}{n-1}\left[\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}}-1\right] \text{ foot-pounds (theoretical)}$$

$$=\frac{144\times1.406\times14.7\times6.10}{0.406}\left[\left(\frac{89.7}{14.7}\right)^{0.29}-1\right]=30{,}832 \text{ ft.-lb.}$$

Adding 15 per cent. for friction we have

$$W_n=35{,}457 \text{ ft.-lb.}$$

If the compression is to be accomplished by a steam engine cutting off at 1/4 stroke the number of cubic feet of 75 lb. steam required to do the work is found as follows:

$$W_n=144\,P_mV_1$$

in which, W_n = work in foot-pounds = 35,457
P_m = mean effective steam pressure in pounds per square inch = 37.8
V_1 = volume of steam in cubic feet after expansion = four times the volume we wish to ascertain.

Introducing values $35{,}457=144\times37.8V_1$

whence $$V_1=\frac{35{,}457}{144\times37.8}=6.52$$

Dividing by 4 $$\frac{6.52}{4}=1.63$$

Adding 15 per cent. $1.63+\frac{1.63}{100}\times15=1.87$ cu. ft. of 75 lb. steam, or practically 2 cu. ft.

In other words, to compress a certain mass of air by steam pressure so as to furnish 1 cu. ft. of compressed air at 75 lb. gage and

at 60° Fahr., requires practically 2 cu. ft. of steam at 75 lb. Now, 1 cu. ft. of steam at 75 lb. weighs $\frac{1}{4.86}=0.206$ lb. (Kinealy steam engine). Total heat required to make 1 lb. of steam at 75 lb. from water having a temperature of 60° Fahr., is:

$$1179-(60-32)=1151 \text{ B.T.U.'s (Kinealy).}$$

Total heat required to make 2 cu. ft. of 75 lb. steam, is:

$$2\times 1151\times 0.206=474 \text{ B.T.U.'s}$$

From this it follows that the number of heat units required to produce by steam energy 1 cu. ft. of air at 75 lb. gage and at 60° Fahr. is 474 B.T.U.'s.

The temperature of 1 cu. ft. of air at the end of the pipe line is $(60+461)=521$ degrees absolute. To double the volume at constant pressure, we must double the temperature, that is, the absolute temperature of the 2 cu. ft. of air at 75 lb. gage would be 1042 degrees and the increase in temperature is

$$1042-521=521 \text{ degrees.}$$

The weight of 1 cu. ft. of air at 75 lb. gage and at 60° Fahr. is from Article 24:

$$W_2=2.7077\frac{P_2}{T_1}$$

$$W_2=2.7077\frac{75+14.7}{60+461}=0.466 \text{ lb.}$$

The specific heat of air at constant pressure is 0.2375. Therefore, to raise the temperature of 0.466 lb. of air at constant pressure by 521 degrees requires

$$0.466\times 0.2375\times 521=58 \text{ B.T.U.'s}$$

This shows that to produce an extra cubic foot of air at 75 lb. by a steam-driven compressor would require 474 heat units, whereas by reheating we have at an expenditure of only 58 heat units made 2 cu. ft. out of the original 1 cu. ft. of compressed air.

The additional power cost of reheating, expressed in heat units, in this case is therefore only one-eighth of that of compression by steam energy.

In practice, no attempt is made to double the volume of compressed air, after it arrives at the air engine, because at tempera-

tures much above 300° Fahr. the lubricant in the motor is apt to charr, causing severe cutting action on the valves, rods, and stuffing boxes.

To increase the volume of the compressed air by reheating from 40 to 50 per cent. is considered quite satisfactory. Beyond that, the air is heated only sufficiently to compensate for heat loss during its passage from the heater to the air engine. To minimize this loss, the heater should be placed as near the point of use as circumstances will permit, and the pipe between the heater and the machine should be well covered.

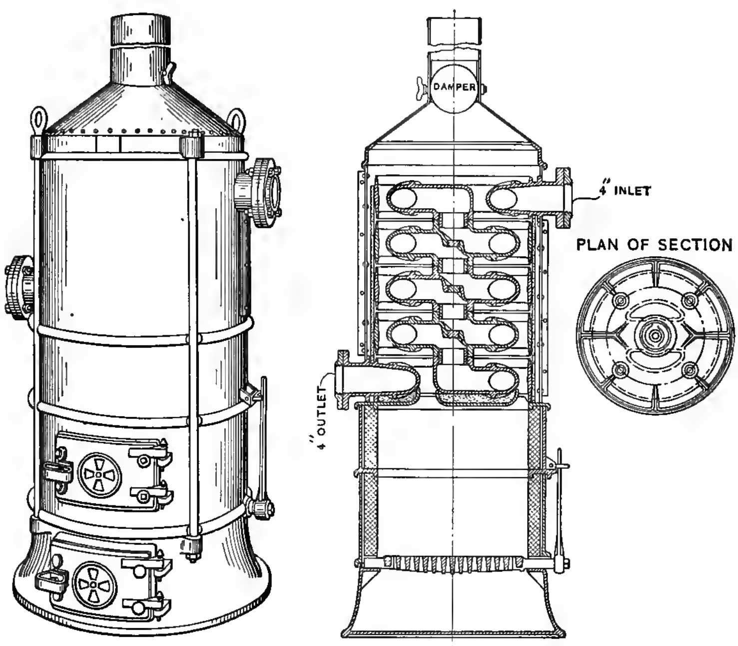


Fig. 19.—Sullivan Air Reheater.

In the following articles are illustrated and described two types of reheaters which are being used in operations where compressed air is employed for power purposes.

AIR REHEATERS

123. The Sullivan air reheater, illustrated in Fig. 19, consists of a series of hollow annular rings, forming the heating surface,

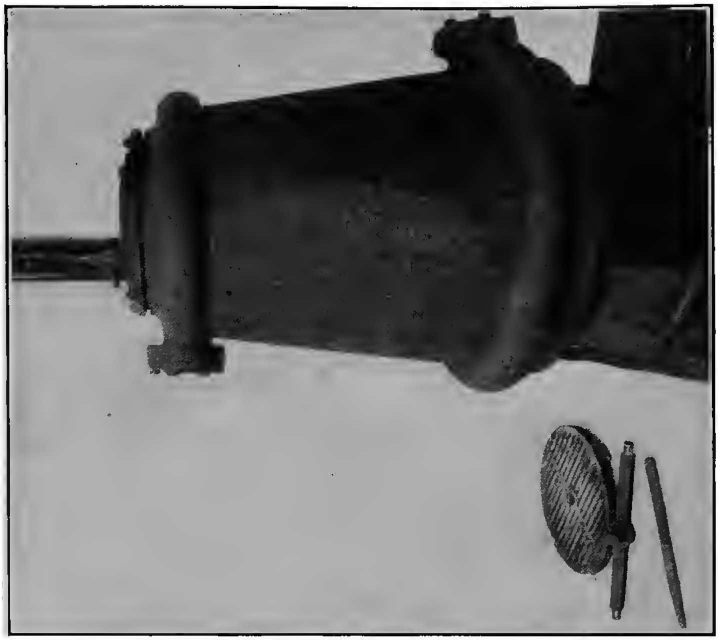


Fig. 20A.

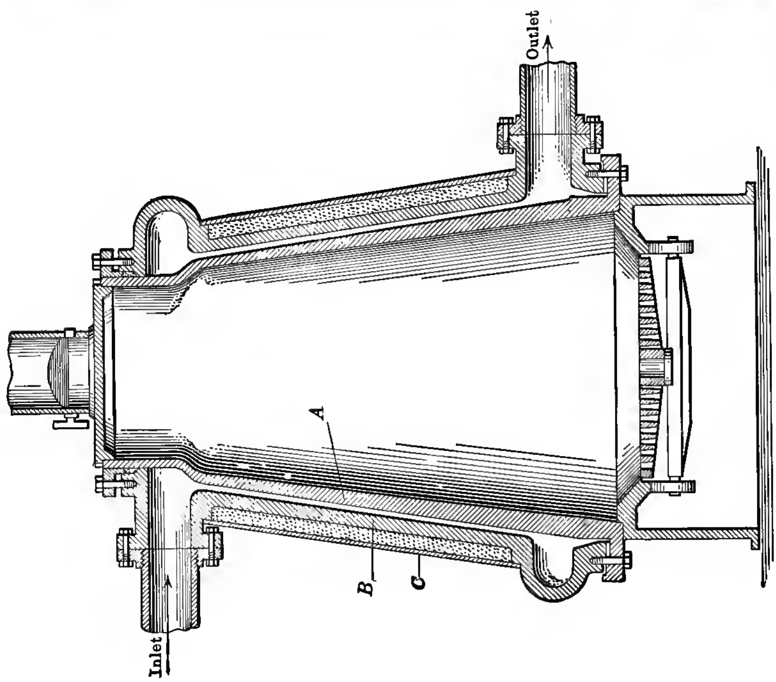


Fig. 20.

Sergeant Air Reheater.

surrounded by asbestos matting and enclosed in a sheet-steel shell. The rings and shell rest upon two cast-iron rings, lined with fire brick, forming the fire box. The latter is provided with dumping grate and doors as shown. The hot gases after circulating around the rings, escape through the hood and smoke pipe on top.

Air enters the reheater at the top and is forced to take a circuitous passage through the annular rings by means of baffle plates so that it comes in contact with the heating surface. The heated sections are designed so as to prevent leakage in the joints, due to expansion. The heated air leaves the reheater by a flanged opening in the bottom ring, similar to that by which it enters the top ring.

These reheaters are designed for burning coal, but may be adapted for oil fuel.

124. The Sergeant air reheater, made by the Ingersoll-Rand company, is shown in Fig. 20. The air enters at the top of the heater, is forced in thin sheets through the annular space between the inner shell (*A*) and outer shell (*B*) of the heater, and leaves the latter at the bottom. The increased air space between intake and discharge pipes, due to the conical shape of the castings, provides for the expansion of the air in heating. The outer shell is surrounded by a mantel (*C*) of sheet iron and the space between the latter and the shell (*B*) is packed with asbestos.

125. Other reheaters, using steam, are successfully employed for surface work. Those described are, in general, not suitable for underground work in mining operations where the smoke from coal or oil fuel is objectionable. Although a number of appliances have been tried for such work, no heater that could be used satisfactorily under all conditions has made its appearance as yet.

PART IV

AIR COMPRESSORS AND ACCESSORIES

CHAPTER XVII

EXAMPLES OF MODERN AIR-COMPRESSORS OF THE RECIPROCATING TYPE

126. In the following articles a few prominent types of compressors, selected at random, are illustrated and described for the purpose of demonstrating the practical application of the theoretical principles discussed in preceding chapters.

The design and construction of compressors is a subject of mechanical engineering. No attempt has been made here to treat this subject in detail. But the writer believes that a few general remarks on the construction of modern compressors will prove helpful to the engineer in making a judicious selection of machines, when called upon to install a compressed-air plant.

127. Fig. **21** gives plan and elevation of five types of steam-driven compressors, showing some, but by no means all the possible combinations of steam and air cylinders. Compressor builders, as a rule, designate them either as "straight-line" or "duplex" compressors.

OPERATION OF STEAM-DRIVEN, STRAIGHT-LINE COMPRESSORS

128. If we study the section of the steam and air cylinders of a compressor as shown in Fig. 22 and assume the piston to move in the direction of the arrow, we note the following conditions:

In the air cylinder, at the beginning of the stroke the resistance to the advance of the piston is practically zero. The pressure, however, begins to rise at once, steadily increasing the corresponding resistance against the piston until close to the end of the stroke at A the receiver pressure is reached, when the discharge valves open. From this point to the end of the cylinder at B the piston travels against a practically uniform maximum pressure in delivering the air into the receiver.

On the return stroke, the compressor being double-acting, the resistance is again zero at the beginning of the stroke and maximum for the latter part of it.

In the steam cylinder the development of power is precisely the reverse of the distribution of resistance in the air cylinder.

Here the pressure is maximum at the beginning of the stroke and practically uniform until cut-off occurs at *D*. Then the

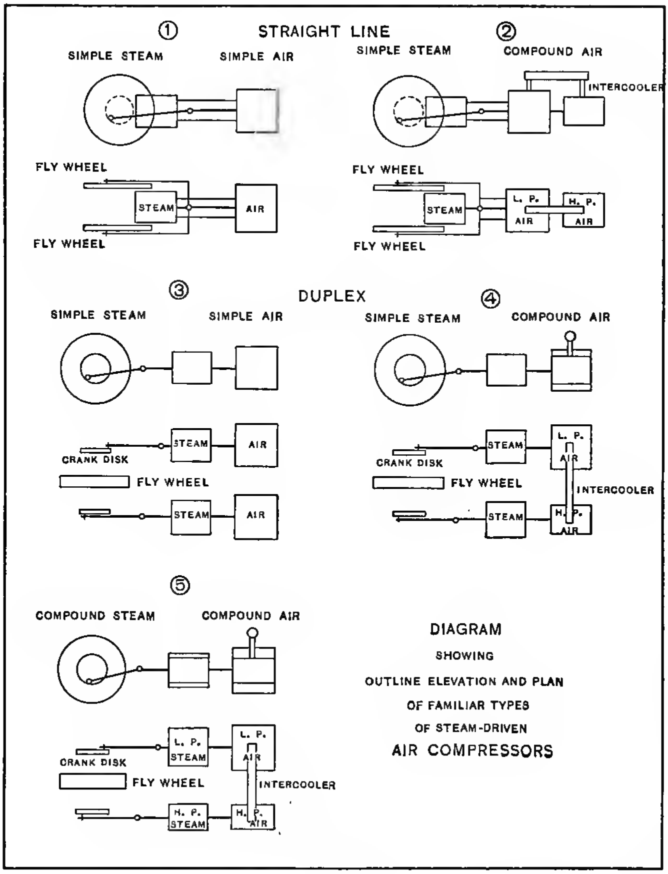


FIG. 21.

pressure rapidly falls all the way to the end; so that in any compressor of the straight-line type the steam power is in excess of the work to be done at the beginning of the stroke in either direction and inadequate to overcome the resistance of the air at the other end of the stroke, except with the assistance of fly-wheels.

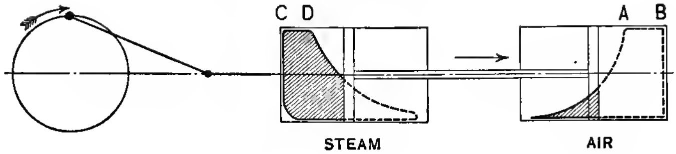


FIG. 22.

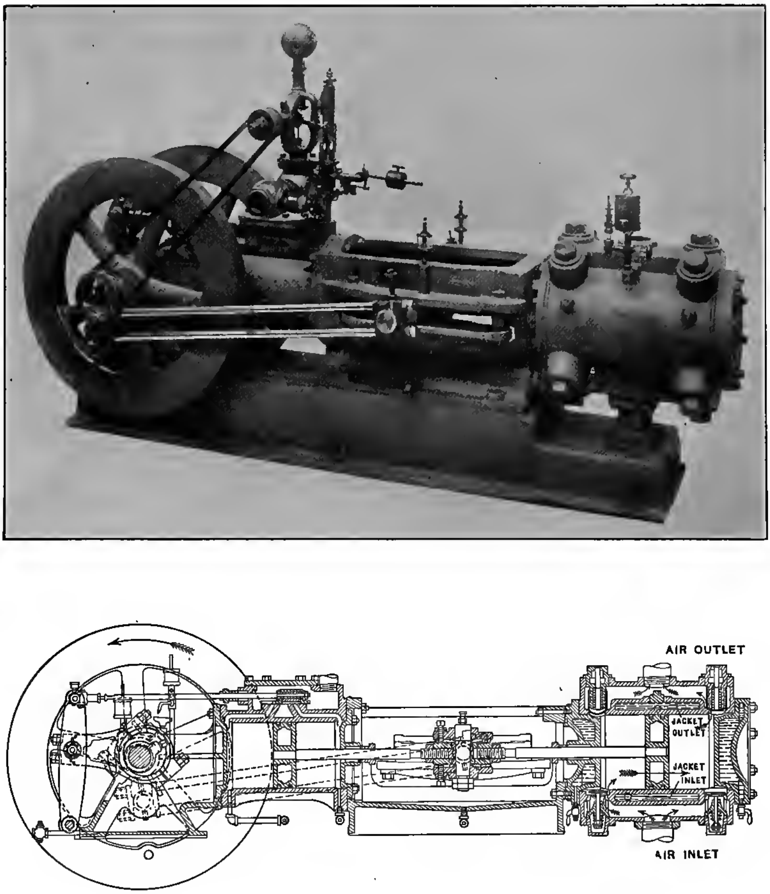


FIG. 23.—Sullivan Straight-line Steam-driven Single-stage Compressor.

The excess pressure at the beginning of the stroke causes the fly-wheels to acquire momentum which they give off at the end of the stroke to overcome the excess of the air-piston resistance.

This unequal application of power to resistance prevents smooth running in this class of compressors and causes severe strains in the moving parts.

129. Type (1). Straight-line, Steam-driven, Single-stage Compressor.—Fig. 23 illustrates a compressor of this type, built by the Sullivan Machinery Company. These machines have one steam and one air cylinder, set tandem on a common piston rod, and two fly-wheels, usually driven by outside connecting rods from a cross-head, which slides between guide plates connecting the steam and air cylinders. They are built and used for pressures up to 90 lb.

Advantages.—Compressors of this type are self-contained, simple in construction, strong and compact, and of moderate price. Compared with duplex machines of equal capacity they occupy a smaller floor space and do not require as expensive foundations as the latter.

Disadvantages.—When running below a certain speed, most straight-line compressors have a tendency to stick on centers. Hence an early cut-off in the steam cylinder is not possible. Such compressors usually run with 5/8 to 3/4 cut-off, resulting in high-steam consumption, averaging from 40 to 50 lb. per horsepower hour.

130. Type (2). Straight-line, Steam-driven, Two-stage Compressor.—Fig. 24 illustrates a machine of this type, built by the Sullivan Machinery Company. These machines have one steam and two air cylinders, set tandem on a common piston rod. The air is compressed in the larger (low-pressure) cylinder to an intermediate pressure, whence it passes by way of an inter-cooler into the smaller (high-pressure) cylinder, where it is compressed to the final pressure.

Advantages.—If properly designed and cared for, these machines have the advantages pertaining to compound compression as pointed out under Article 75.

Disadvantages.—In addition to the disadvantages pointed out for Type (1), which apply to all straight-line compressors, the compounding of the air cylinders complicates the machine, increases the relative cost of it for the work it does, makes all

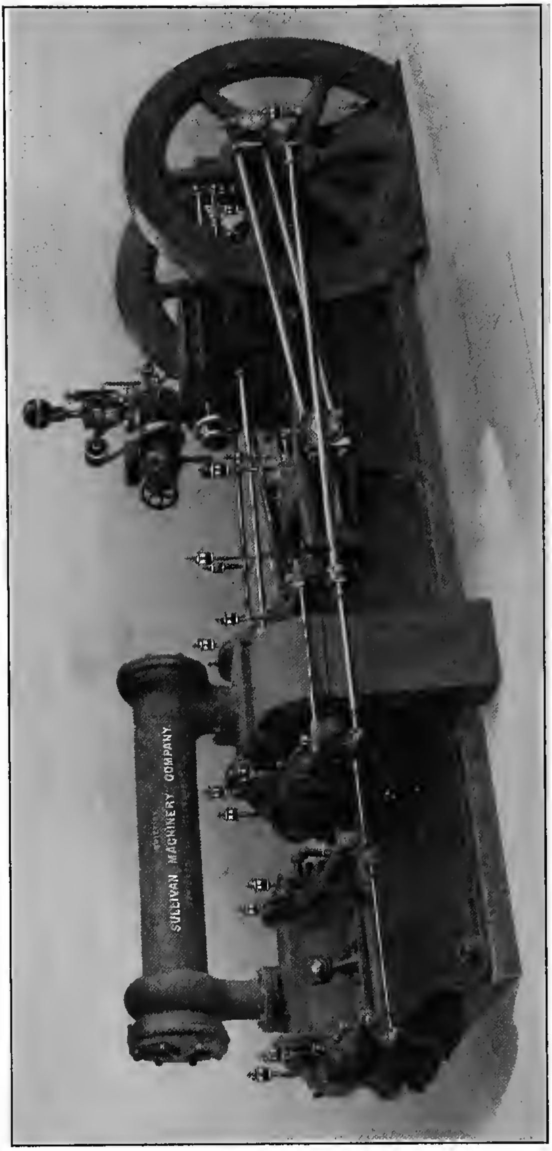
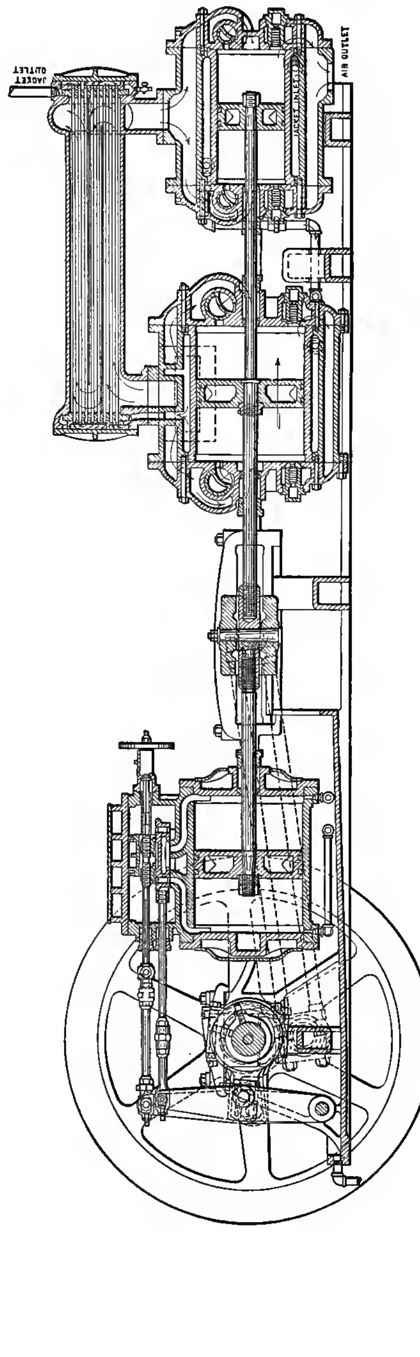


Fig. 24.—Sullivan Sraight-line Steam-driven Two-stage Compressor.



SECTIONAL VIEW OF STEAM CYLINDER

SECTIONAL VIEW OF AIR CYLINDERS

Sullivan Straight-line Steam-driven Two-stage Compressor.

parts less accessible for adjustment, while it has left the machine with its usual inability to run at slow speed.

131. Operation of Steam-driven, Duplex Compressors.—A duplex air compressor is, in essential effect, a combination of two straight-line machines, so far as the steam and the air cylinders are concerned, with a single crank shaft and a single fly-wheel serving both, there being a single connecting rod for each side and a single crank on each end of the shaft.

In the duplex compressor the operating conditions are in decided contrast to those described for the straight-line compressors.

Quartering Cranks.—The first special feature of advantage of the duplex machine is in the arrangement of the cranks in relation to each other upon the ends of the shaft. These are set with one of the cranks a quarter of a circle in advance of the other, the result of which is to so time the movements of the pistons on the two sides of the machine that one will be at nearly midstroke when the other is at the beginning or end of its stroke. The two sides thus alternately help each other over the hard places, and, while not under nearly as great obligation to the fly-wheel, their

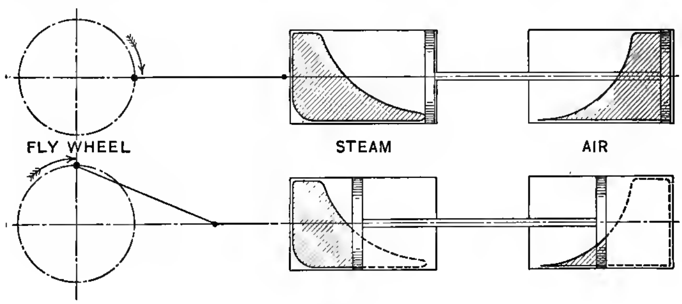


FIG. 25.

action is much steadier and so free from excesses of pressure over resistance or of resistance over pressure, that the rotation is more uniform. The practical limit of speed is lowered to perhaps one-quarter of the lowest speed permissible in the straight-line type, so that, if the cut-off on the steam cylinder is properly set, the machine may be made to automatically stop and start itself and to run at any speed down to the lowest, as the air consumption may require. Waste of power and steam, consequent upon the necessity of running at high speed to prevent centering,

will be avoided. The conditions of pressure and resistance are illustrated graphically in the diagram Fig. 25.

The two steam and air cylinders of the compressor are shown in the diagram one above the other. In the two upper cylinders minimum power in the steam cylinder is being applied to maximum resistance in the air cylinder at the end of the stroke. In the two lower cylinders excess power is being applied to small resistance at midstroke, the surplus pressure acting to carry the compressor past the center of the upper cylinders.

Similar conditions can be shown for other positions of the pistons.

131a. Compared with straight-line machines, duplex compressors offer several disadvantages which should be taken into consideration when planning an installation.

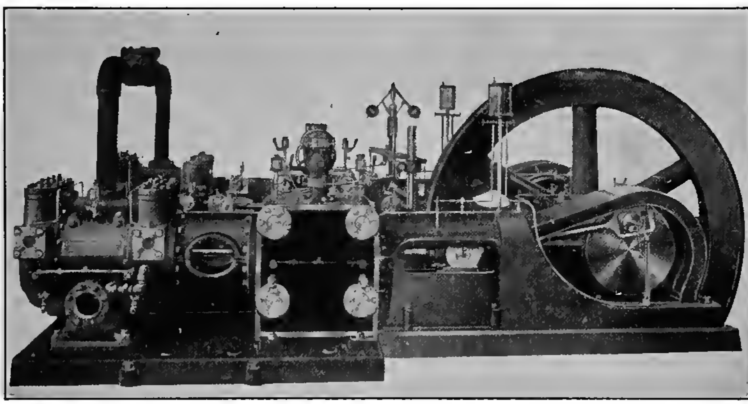


Fig. 26.—Laidlaw-Dunn-Gordon Duplex, Steam-driven, Single-stage Compressor.

For any given output of air they are more expensive in first cost and up-keep, for there is double the machinery. There are double the chances of delays, for either side may be necessarily stopped and then all the air is shut off until adjustments can be made to both machines. A heated journal on either side will stop both. The friction of the duplex machines exceeds on an average by about 5 per cent. the friction of two machines working separately.

All duplex compressors occupy much more floor space than straight-line machines of the same capacity, consequently require larger and more costly buildings and foundations.

132. Type (3). Duplex, Steam-driven, Single-stage Compressor.—Fig. 26 shows a machine of this type, built by the

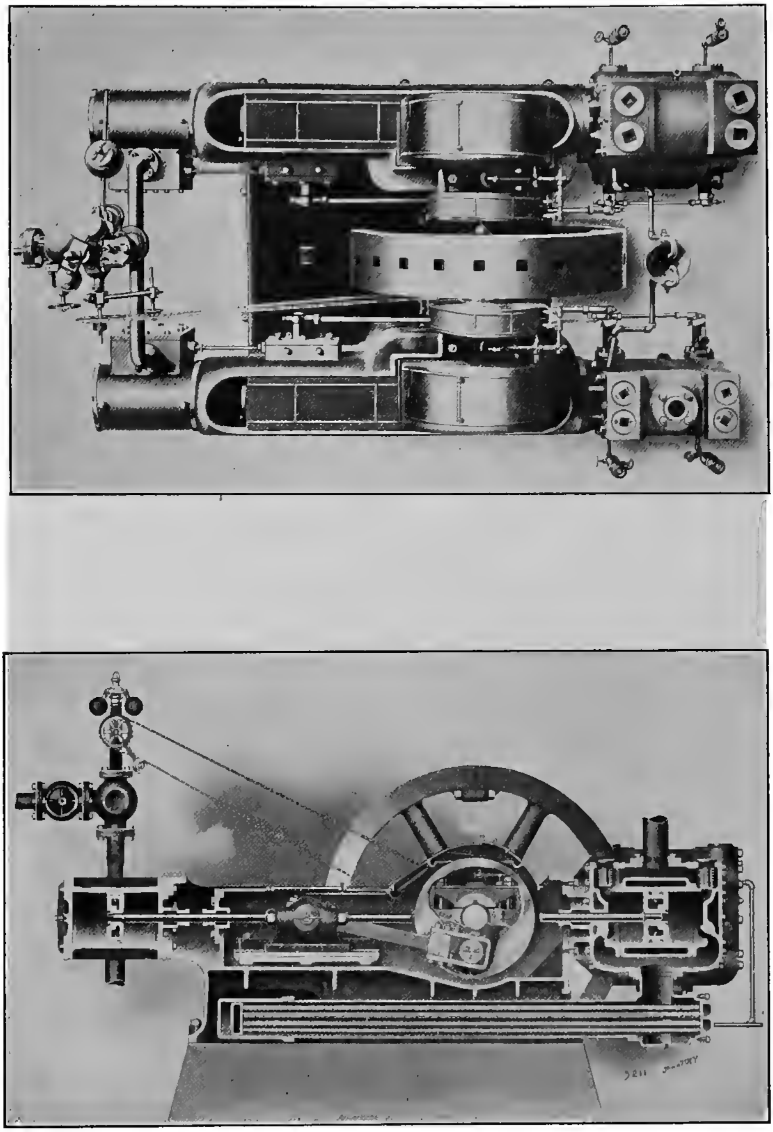


FIG. 27.—Ingersoll-Rand Duplex Steam-driven Two-stage Compressor.

Laidlaw-Dunn-Gordon Co. It is merely a combination of two straight-line, single-stage compressors of type (1), set on the same shaft with one fly-wheel instead of two.

It combines all the advantages and disadvantages of a duplex compressor, pointed out in Articles 131 and 131a. Compared with types 4 and 5, it has the advantage that, if necessary, one side can be operated as a complete machine.

133. Type (4). Duplex, Steam-driven, Two-stage Compressor.—Fig. 27 illustrates a compressor of this type built by the Ingersoll-Rand Co. It has simple steam cylinders and cross-compound air cylinders. The inlet valves of both the low- and high-pressure air cylinders are of the Corliss type. The inter-cooler is placed in the cast-iron frame, which makes the compressor more compact.

Compressors of this type partake of all the advantages and disadvantages of the duplex feature as well as of stage-compression, as pointed out in Article 75. One objection to cross-compound air cylinders in duplex machines is that under no circumstances can one side be operated as a complete machine.

134. Type (5). Duplex, Steam-driven, Two-stage Compressor.—Fig. 28 illustrates a compressor of this type, built by the Allis-Chalmers Company, with the inter-cooler removed. Both air and steam cylinders are cross-compound. Inlet valves are of the Corliss type. It is not possible to operate one side of this compressor as a complete machine, on account of the cross-compound feature, which requires the operation of both sides at the same time.

135. Other Types of Steam-driven Compressors.—For illustration of other types and combinations of steam-driven, single- and multi-stage compressors, the reader is referred to the catalogues and bulletins of manufacturers which, besides copious illustrations, usually contain a large amount of useful data on air compression. The remarks contained in the preceding articles should enable the reader to draw fairly correct conclusions as to the merits of the one or the other type and make of compressor when referred to the needs of any contemplated installation.

POWER-DRIVEN AIR COMPRESSORS

136. A large class of compressors used in the various industries, are of the power-driven types. That is, they consist of one or more air cylinders, the piston rod of which is connected through

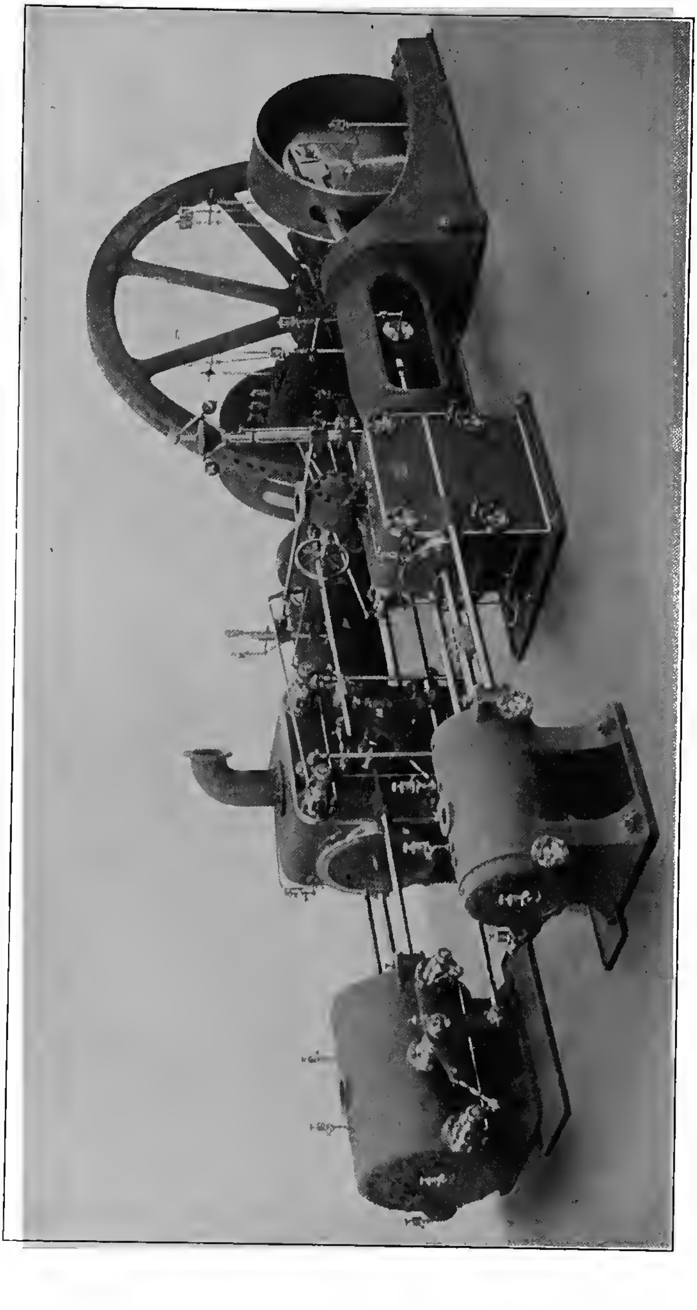


FIG. 28.—Allis-Chalmers Duplex Steam-driven Two-stage Compressor.

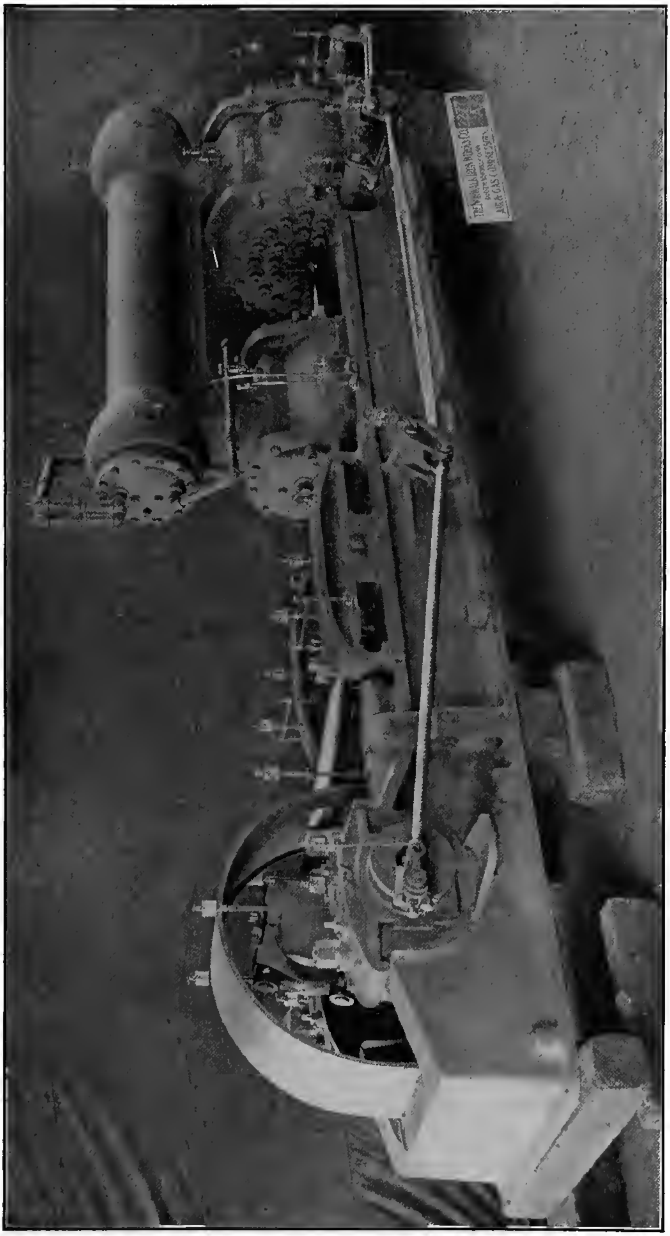


FIG. 29.—Norwalk Belt-driven Two-stage Compressor.

a connecting rod and crank to a revolving shaft, the latter being driven from a central power plant, or by a water-wheel, or by an electric motor.

In selecting a power-driven compressor, it must be borne in mind that it cannot be hurried, neither can it be run at a speed little less than the maximum. Steam-driven machines can be run at variable speed to suit the requirements, but the power-driven compressor must always run at full speed, and variations of demand can only be met by unloading, either wholly or in part as circumstances may require.

For unloading devices see Articles 156–160.

Compressors of small power can be driven by belts, chains or gears. Moderately large powers, unless driven direct, are dependent upon ropes or belts, while for compressors of very large capacity direct drive seems the most satisfactory.

BELTED, COMPARED WITH DIRECT STEAM POWER COMPRESSORS

137. The question is frequently asked: Under what conditions is a belted compressor more advisable than one having its own independent steam engine? In an establishment having a large high-class main engine of abundant power, the belt pattern offers the advantage of compressing the air with the same steam econ-

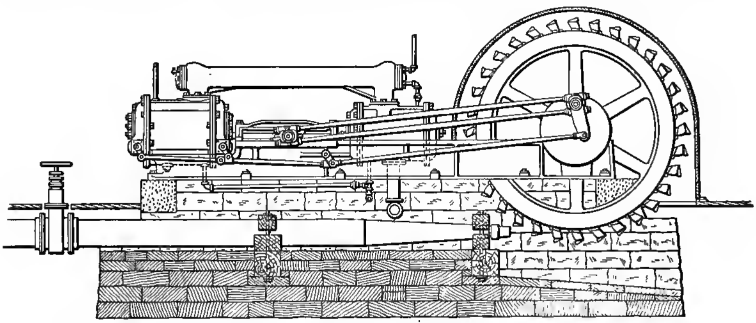


Fig. 30.—Norwalk Two-stage Compressor with Water-wheel Drive.

omy as is obtained in the large steam engine, and will therefore prove more economical in first cost as well as in operation. Economy, however, must always be studied not in the engine alone, but in the compressor as well and when the air demands are of considerable relative consequence, the power required may necessitate the individual engine for the compressor.

138. Fig. 29 illustrates a two-stage compressor of the belt pattern, built by the Norwalk Iron Company. The large band wheel serves for a driving pulley as well as for a fly-wheel and can be belted to the pulley on a large main drive shaft or to the pulley of an electric motor.

139. Compressors with Direct Water-wheel Drive.—Fig. 30 illustrates a compressor of this type built by the Norwalk Iron Company. For the band wheel a heavy fly-wheel is substituted and on this the water-wheel buckets are mounted.

Where abundant water power is available for continuous service such compressors may be used to great advantage.

ELECTRICALLY OPERATED COMPRESSORS

140. Like all power-driven compressors, electrically operated compressors must be run at constant speed and must therefore

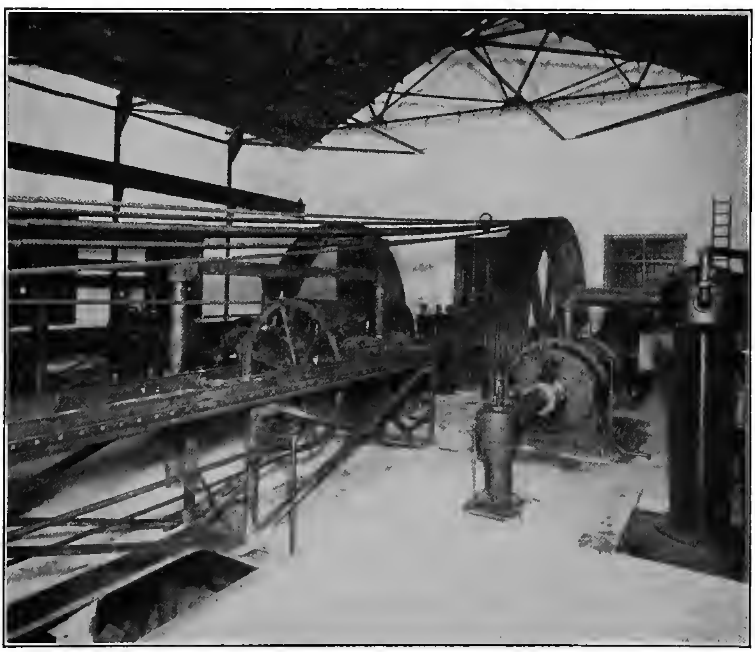


Fig. 31.—Ingersoll-Sergeant Rope-driven Compressor.

be provided with unloading devices to regulate the output, whe used for intermittent demand.

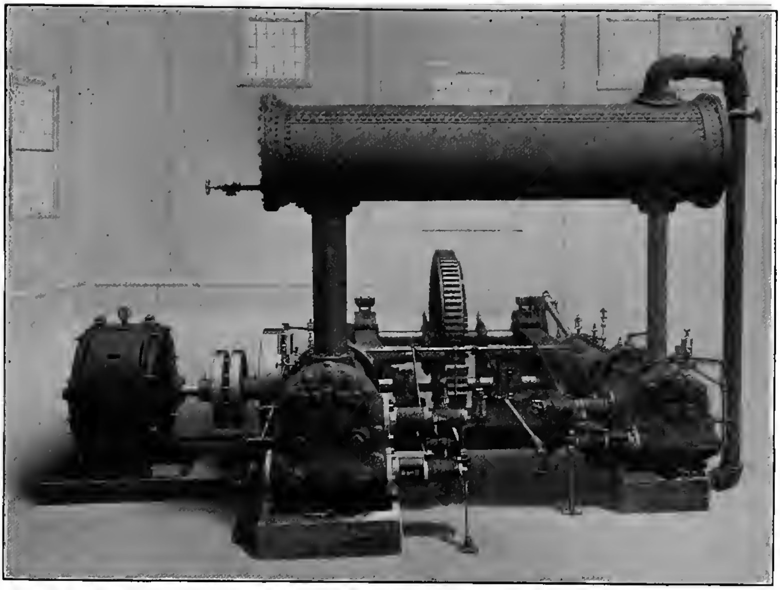


FIG. 32.—Nordberg Electrically-driven Geared Two-stage Compressor of the Duplex Type.

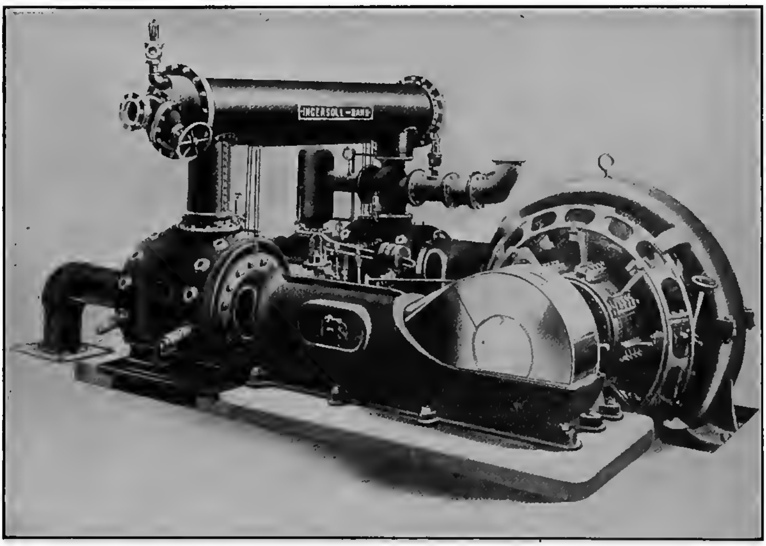


FIG. 33.—Ingersoll-Rand Direct-connected Electrically-driven Two-stage Compressor.

Besides belt or rope driven, in which the pulley of the motor is connected by belt or ropes with the band wheel of the compressor, as shown in Fig. 31, electric motors are also geared or direct connected to the revolving shaft of the compressor.

141. Fig. 32 illustrates an electrically driven, geared, two-stage compressor of the duplex type, built by the Nordberg Mfg. Co. The inlet valves which are of the Corliss type are released by an unloading device from the operating mechanism, when they are wide open and are kept in that position until more air is required. The releasing is effected by cams, operated by a frictionless plunger on which the air pressure acts in opposition to a weight. These cams throw out a latch so placed on the valve operating lever, that it closes the valve, while the opening is effected by a projection acting on the valve operating lever. The cams are so adjusted that first one and then the other engages the releasing latch.

142. Fig. 33 illustrates a direct-connected, electrically driven, two-stage compressor, built by the Ingersoll-Rand Company.

CHAPTER XVIII

IMPORTANT MECHANICAL FEATURES OF AIR COMPRESSORS

143. Without going into detail of construction, attention will be called to some of the mechanical features which influence the operation of a compressor to such an extent that unsatisfactory results can in most cases be traced to either defective construction or to neglect of proper care of certain parts of the machine. A proper understanding of their function will enable the operator to trace to the most probable source, any failure of the compressor to do its duty, and to apply the necessary remedies, if such failure is not due to inherent imperfections of the machine itself.

INLET VALVES

144. The inlet valves of a compressor are either of the poppet type, being held to their seats by springs, or they are mechanically moved, resembling in their general form and operation the steam valves of a Corliss engine.

145. Poppet Inlet Valves.—All poppet valves, whether used as inlet or discharge valves, consist essentially of three main parts: a valve proper, a valve guide and a spring. For illustration see Fig. 36.

In general, poppet inlet valves are open to the objection that, inasmuch as the springs must insure prompt closing at all speeds, they must have considerable strength. This causes throttling of the inlet and hence loss of volumetric efficiency and requires extra power to make up for this loss.

Another objection is, that the incoming air passes in a very thin stream over these heated surfaces and is itself heated and rarefied as a consequence. The undesirable effects of these conditions have been pointed out under Articles 54 and 87.

These objections have led to the introduction of inlet valves of the Corliss type.

146. Inlet Valves of the Corliss Type.—Fig. 34 illustrates the construction of an air inlet valve of the Corliss type, employed in some of the Ingersoll-Rand compressors. (See Fig. 27.)

The valve is made of cast iron and is operated by a steel stem "*A*," which has a large flange "*B*" provided with a series of tongues on its inner face, machined to match with grooves on end of valve "*C*." Valve bonnet "*D*" is of the stufferless type, the stem being made self packing by means of a fiber washer "*E*." Contact is maintained between faces of fiber washer and face of valve stem collar, also bonnet face, by means of a spring and thimble "*F*" in back bonnet "*G*." Lubrication for the valve is provided for at "*H*" and "*I*."

Valves of the Corliss type are positively moved from the main shaft of the compressor as shown, for instance, on the compressor illustrated in Fig. 28.

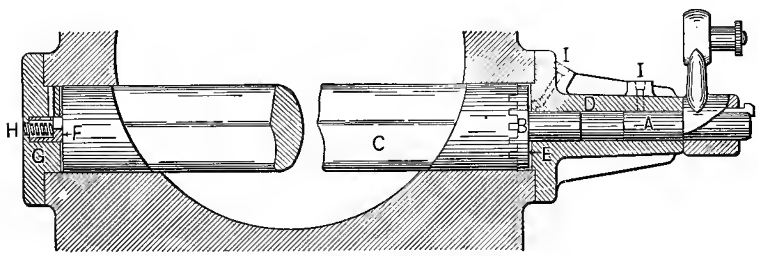


Fig. 34.—Air Valve of the Corliss Type.

The large port of these valves and their positive operation insures an air supply that keeps pace with the speed of the machine.

147. Ingersoll-Rogler Valve.—The valve illustrated in Fig. 35 is of the disc type. It consists essentially of a valve seat (*A*), cast with circular ports, which supports valve (*F*) and cushion plate (*H*) that are separated from each other by washers (*E* and *G*). All are held together by valve bolt (*B*). Portions (*M*) of valve (*F*) are elastic spring arms that hold the valve absolutely in one position and always seat it in the same place. The four spring arms of cushion plate (*H*) hold the valve on its seat against a slight tension of the integral valve arms (*M*). As soon as the proper air pressure is reached, the valve opens against these springs and closes again at the instant the piston starts on its return stroke.

Owing to the light weight and low lift, the valve is subject to little wear and shock. The time required to open and close it being very brief, higher speeds and therefore greater capacity are possible than would be safe with other types of valves.

The absence of gears for operating the valve eliminates friction and since there are no rubbing parts, the valve needs no lubrication.

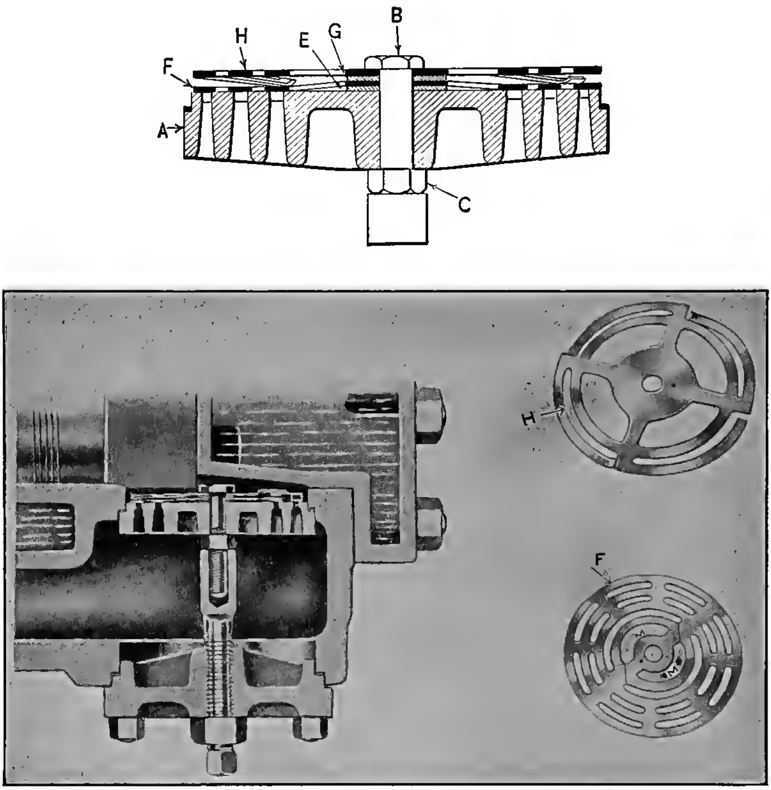


FIG. 35.—Section Showing Inlet Valve in the Air Cylinder of a Modern Ingersoll-Rand Compressor.

DISCHARGE VALVES

148. Discharge valves, like inlet valves, are either of the poppet or disc type or they are mechanically moved and of similar construction as the inlet valve shown in Fig. 34.

149. Poppet Valves.—Fig. 36 illustrates a simple construction of a poppet discharge valve, used in some Ingersoll-Rand compressors. (See Fig. 10.)

The valve proper is ground to an accurate seat, while the cap or valve-guide is ground to a wide guide surface insuring the re-

turn of the valve to its seat with precision and tightness. A small volume of air is compressed between valve and guide at the end of the lift, affording a cushion which removes shock without interfering with the quick action of the valve. The valve is free to turn and is self-grinding. The spiral spring is made of the proper pitch and strength to return the valve to its seat at the proper moment.

Discharge valves of the poppet type are open to similar objections as poppet inlet valves, which has led to the introduction of mechanically moved valves of the Corliss and other types.

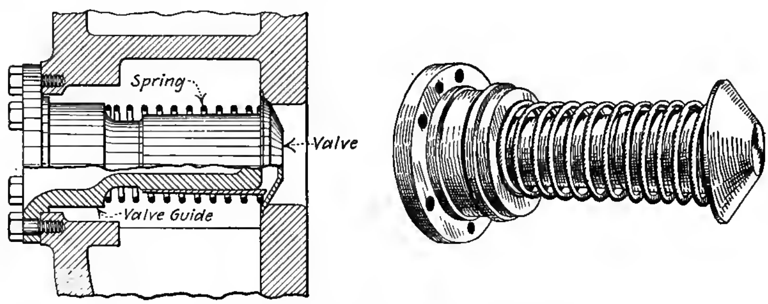


Fig. 36.—Air Valve of the Poppet Type.

150. Mechanically Moved Discharge Valves.—When of the Corliss type, they are essentially of the same construction and are operated in the same manner as the valve illustrated in Fig. 34.

The principal objection to positively operated discharge valves is, that the point of opening is fixed and thus too late when the discharge pressure is below, or too early when above normal pressure, as this frequently happens with compressors supplying an intermittent demand. Such compressors, when using mechanically controlled discharge valves, have the latter usually arranged so that they are free to open automatically, but are positively closed.

151. Fig. 37 shows such a valve, employed in some compressors built by the Allis Chalmers Co. As seen, the inlet valves are of the usual Corliss type; the discharge valves (A) open as soon as the air pressure in the cylinder reaches that of the air in the receiver and are positively closed by plungers (B), which are operated by being connected to a wrist plate driven by an eccen-

tric on the main shaft. The movement of the plunger is so timed as to positively bring the valves to their seat just as the piston reaches the end of the stroke, thus avoiding any slip of the air back by the valves. During the return stroke of the piston the valves are held to their seats by the discharge air pressure until the process is repeated on the succeeding forward stroke. In closing, the air between plunger and valve forms a cushion so that the valve is brought to its seat without noise or pounding.

152. Fig. 38 shows an arrangement used in some compressors, built by the Nordberg Mfg. Co. Both inlet and discharge valves

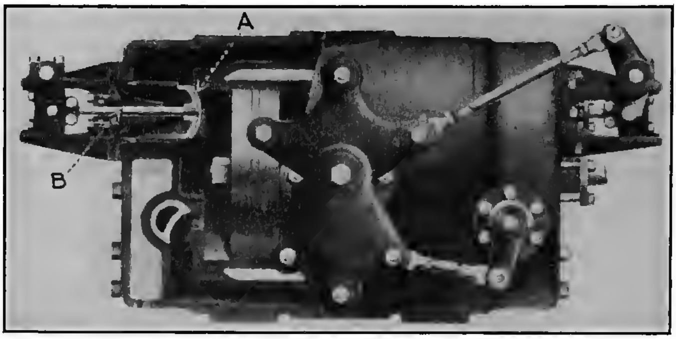


Fig. 37.—Allis-Chalmers Discharge Valve.

are of the Corliss type. In the center of each discharge valve are fitted a row of self-acting poppet valves, which open automatically when for any reason the discharge pressure is below normal.

153. Corliss valves are not suited for discharge valves in single-stage compressors, compressing to more than 30 lb. gage, because the time between the opening and closing is too short to be performed by a positive mechanism. In such cases self-acting poppet valves are used.

They may be used, however, in single-, two- or three-stage compressors, in which they have to be kept open during nearly one-half of the stroke.

THE INTER-COOLER

154. Inasmuch as the intercooler is the principal device by which a saving of power in stage compression is accomplished (see Article 57), it must be planned and designed so as to cool to initial temperature the heated air that passes through it on its way from one cylinder of a compressor to another.

To do this effectively, it must possess the following essential properties:

1. The cooling surface offered to the circulating air must be

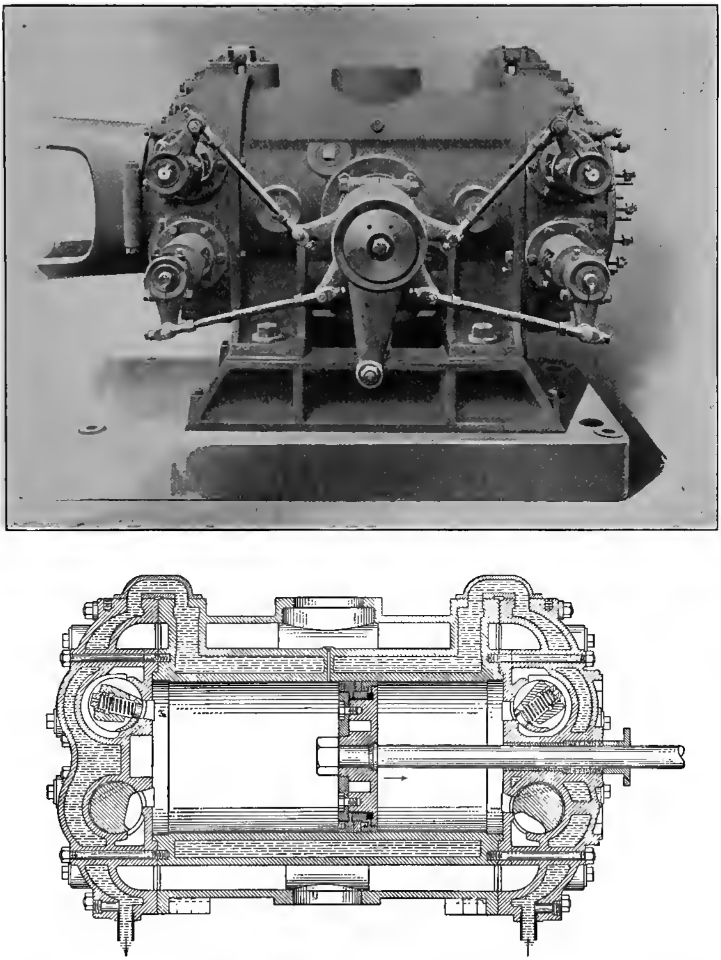


FIG. 38.—A Nordberg Discharge Valve of the Corliss Type Fitted with Self-acting Poppet Valves.

ample. This is generally based on the quantity of "free air" compressed per minute.

2. The total volume should be as large as possible. Of two inter-coolers having the same amount of cooling surfaces, the

one of larger volume offers the advantage of allowing the air to be in contact with the cooling tubes for a longer period.

3. The water-circulation should be planned so as to make the water flow unrestricted and with proper velocity through the pipes and thus absorb and carry away the maximum number of thermal units contained in the air.

4. It should be provided with means for bringing the air in continuous contact with the cooling surfaces. This is usually accomplished by so-called "baffle plates."

5. It should have convenient appliances for draining the condensed moisture, and should permit easy access for inspection and repairs.

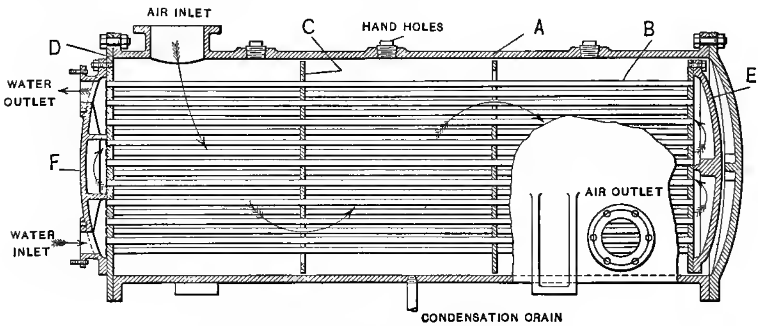


FIG. 39.—Section of Intercooler.

155. Fig. 39 gives a sectional view of an inter-cooler built in accordance with modern practice. It consists of a cylindrical iron shell "*A*" containing a nest of tubes "*B*," through which cold water is circulated. The tubes are so spaced as to divide the air into thin sheets, and by means of baffle plates "*C*," the air is deflected and brought in contact with all parts of the cooling surface, before it leaves the inter-cooler. The tubes are expanded into tube sheets "*D*," and the rear tube sheet is covered by a head "*E*," which is in no wise connected to the shell but is free to slide within it, thus providing for any differences in expansion between shell and tubes. The rear end of the shell is closed by a separate head. The front head "*F*" can be removed and the tubes withdrawn for inspection and cleaning. The heads are provided on the inside with ribs, which abut against the tube sheets and compel the water to pass from end to end of the intercooler several times, thus obtaining the maximum cooling effect from a given quantity of circulating water.

CHAPTER XIX

COMPRESSOR ACCESSORIES

156. The most essential accessories of a compressor plant are automatic regulators and receivers. Only a few types of each are illustrated and described in the following articles. They were selected at random, not with any intention of giving them preferences over other designs or makes, but merely to demonstrate how certain demands which are made on almost every compressor plant may be filled by mechanical means.

AUTOMATIC REGULATORS

157. In the industries using compressed air, particularly in mining operations, the consumption of air is often irregular and intermittent. For short periods it may cease entirely.

To keep on compressing air when the demand is falling off would mean a waste of energy in that the surplus air would simply blow off through the safety valve of the receiver. To prevent such waste, compressors supplying an irregular demand are provided with so-called automatic regulators.

Power-driven compressors, which must run at constant speed, and steam-driven, straight-line compressors which are liable to stick on centers when run below a certain speed, are usually provided with so-called "unloaders."

Steam-driven, duplex compressors may use unloaders or speed governors, or a combination of both.

158. Air Cylinder Unloaders.—These devices are designed to automatically shut off the supply of free air to the compressor when the consumption decreases.

After shutting off the in-take, all the useful work ceases and only sufficient energy is expended to overcome friction of the moving parts.

Fig. 40 shows an unloading device, built by the Union Steam Pump Co. It consists of a casing (*a*) and a plunger (*b*), which controls the admission of air into the compressor through the

inlet pipe (*c*). Attached to one side of the casing is an auxiliary piston (*d*), a lever (*f*) and a weight (*g*).

The air pressure on the auxiliary piston is balanced by the weight which can be adjusted to unload the compressor at any desired pressure. When the decreased demand for air raises the pressure in the receiver beyond the normal, this increased pressure lifts the auxiliary piston (*d*), closes the port (*h*) and admits air at receiver pressure through port (*m*) under the plunger (*b*). The latter is thus raised and closes the air inlet pipe (*c*).

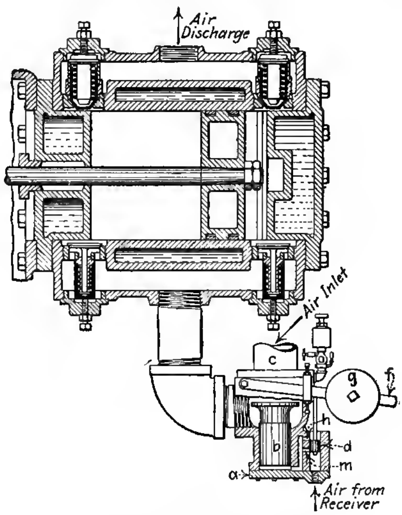


FIG. 40.—Air Cylinder Unloader.

When the receiver pressure falls to normal pressure again, on increased demand of air, piston (*d*) is pressed downward by weight (*g*), port (*m*) is closed, while the air confined under plunger (*b*) is exhausted into the atmosphere through port (*h*). Plunger (*b*) by its own weight drops into its first position, thus opening the main inlet pipe (*c*) and allowing the compressor to resume its useful work.

159. Combined Speed Governor and Air-pressure Regulator.—In mining operations it happens at times that rock drills, hoists, pumps, etc., using air, are all started more or less simultaneously, causing the compressor to run at an injurious speed to supply the unusual demand. At other times the demand may sink below the normal or cease altogether.

Compressors subject to such conditions are usually provided with a combined speed governor and pressure regulator.

160. Fig. 41 shows such a device, furnished with certain compressors of the Ingersoll-Rand Co. This device consists of a regular fly-ball governor (*a*) with an auxiliary air cylinder (*b*) for holding a constant air pressure in the receiver. A casing (*c*) contains a special balanced throttle valve, the spindle of which is connected to the governor, the latter being belt- or chain-driven

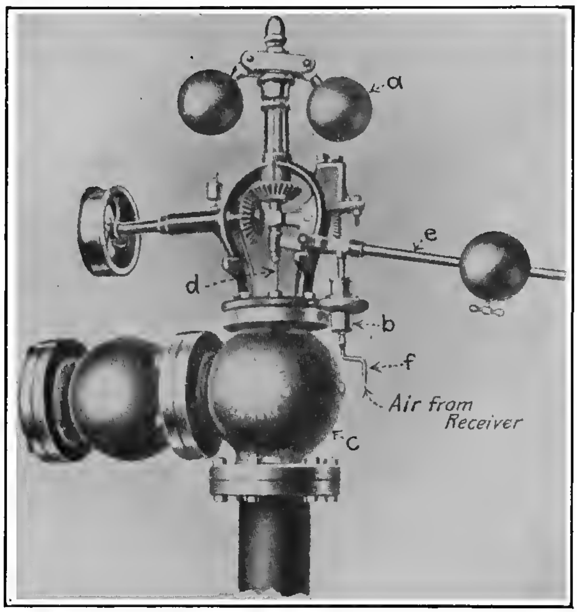


Fig. 41.—Ingersoll-Rand Combined Speed Governor and Air Receiver.

from the compressor shaft. By this arrangement the steam supply is throttled when the speed exceeds the desired limit, which provides a safety stop against "runaways" should an air pipe be broken. The piston in the air cylinder (*b*) presses against a weighted lever (*e*). This cylinder is connected by a small pipe (*f*) to the top of the air receiver. The inner end of the weighted lever connects with the spindle (*d*) of the balanced throttle valve through a link which makes the action of the air cylinder (*b*) independent of the governor (*a*). When the pressure in the receiver exceeds the normal, the weighted lever (*e*) is raised and

the balanced throttle valve closed to a point which admits just enough steam to turn the machine over at the speed necessary to supply a volume of air equal to that drawn from the receiver.

If, from any cause, the air pressure in the receiver diminishes, the weighted lever gradually drops, owing to the decrease of pressure in the small cylinder (*b*). This action opens the throttle admitting more steam into the engine. Should an air pipe break, or should too great a demand be made upon the compressor, keeping the air pressure down so that the air piston does not perform the work, the machine will speed up to a point where the centrifugal governor partially closes the throttle, bringing the engine back to its rated full speed or the speed for which the governor is set.

AIR RECEIVERS

161. Air receivers have become indispensable accessories of every compressed-air installation. Since their design and size influence to a large extent the working of the whole system, the principal functions which they have to perform should be well understood.

Receivers are used for three distinct purposes:

1. To equalize the pulsations of the air coming from the compressor intermittently and to cause it to flow with a uniform velocity into the pipe line. Unless there is ample space for accommodating the air coming from the compressor, the pressure will run up momentarily in excess of the normal, thus throwing unnecessary strain on the machine and consuming extra power.

Receivers are employed to provide this space and in order to perform this function effectively, they should be placed within a few feet of the compressor and connected with it by a pipe of sufficient size.

2. To keep the friction of air in the pipe line as small and as uniform as possible, thereby preventing a loss of energy. In Article 100 it has been shown that friction increases with velocity and the latter increases with the difference of pressure at both terminals of the pipe line. It is therefore important to keep this difference as small and as uniform as possible. In a long line this is best accomplished by placing another receiver at the end of the line, close to the air engine. Just as a receiver near the compressor prevents the rise of pressure above the normal when

air is forced into the pipe, one at the end of the line will prevent a sudden fall of pressure below the normal when air is quickly withdrawn from the pipe line.

3. To collect the moisture and grease which the air carries in suspension and which would otherwise be carried into the pipe line by the force of the current. By allowing the heated air to pause in its flow through the receiver, it is cooled and will therefore drop most of the water and the oil, which at proper intervals are discharged through suitable drain pipes.

A receiver, unless made of prohibitory size, can never act as a reservoir for compressed air, because upon withdrawing air from

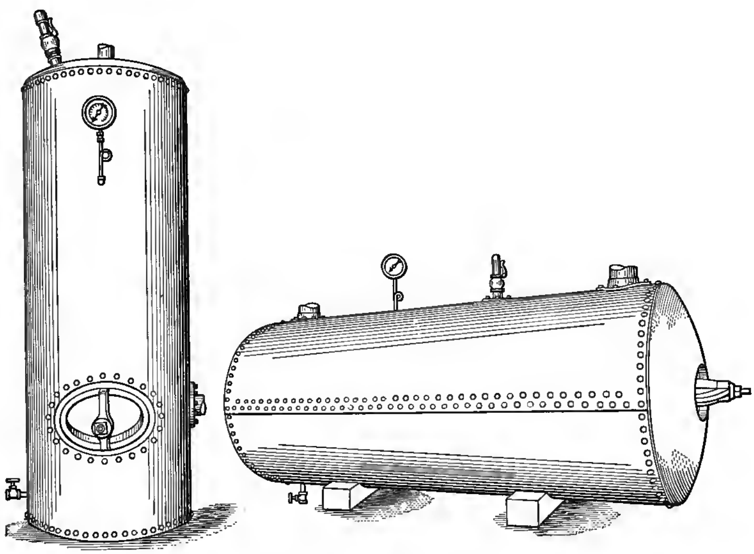


Fig. 42.—Air Receivers.

it, the pressure falls so rapidly that even if of huge dimensions, a receiver could supply the demand only for a few minutes, should the compressor stop for that period of time.

In order to fill the legitimate requirements pointed out above, the dimensions of a receiver must conform to the capacity of the compressor and the discharge pressure of the air. What these dimensions should be for any individual installation is a matter that has been determined largely by experiment.

Compressor manufacturers usually furnish receivers to go with a compressor of given capacity and list in their catalogues the

proper size of a receiver corresponding to the output of the compressor in cubic feet of free air per minute.

Air receivers are built either horizontal or vertical. They are cylindrical vessels made of sheet steel of large tensile strength. The girth seams are single, and the side seams double-riveted. A manhole is provided for inspection and repairs.

Each receiver is usually provided with a pressure gage, a safety valve and a blow-off cock.

Fig. 42 shows a vertical and a horizontal receiver built by the Sullivan Machinery Co.

162. After-coolers.—Inasmuch as they perform practically the same function as an inter-cooler, they are usually of the same or very similar construction as the inter-cooler shown in Fig. 39. They are sometimes employed in place of an ordinary receiver in order to realize more fully the saving of power that results from the partial cooling of the air in the receiver. Such cooling reduces the momentary increase of pressure due to the heat of compression and as a consequence diminishes the power required in forcing the air out of the compressor cylinder into the receiver.

They also secure a more complete cooling of the air before it enters the pipe line, and therefore a more perfect extraction of moisture and grease carried in suspension by the heated air.

APPENDIX.

TABLES I TO IX

TABLE I.—WEIGHT OF 1 CU. FT. OF AIR AT VARIOUS PRESSURES AND TEMPERATURES

Based on an Atmospheric Pressure of 14.7 lb. Absolute at Sea Level

Temperature of air, deg. Fahr.	Gage pressure, pounds																				
	0	5	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
	Weight in pounds per cubic foot																					
−20	.0900	.1205	.1515	.2125	.2744	.3360	.3970	.4580	.5190	.5800	.6410	.702	.7635	.825	.886	.948	1.010	1.165	1.318	1.465	1.625	1.930
−10	.0882	.1184	.1485	.2090	.2685	.3283	.3880	.4478	.5076	.5674	.6272	.687	.747	.807	.868	.928	.989	1.139	1.288	1.438	1.588	1.890
0	.0864	.1160	.1455	.2040	.2630	.3215	.3800	.4385	.4970	.5555	.6140	.672	.731	.790	.849	.908	.968	1.114	1.260	1.406	1.553	1.850
10	.0846	.1136	.1425	.1995	.2568	.3145	.3720	.4292	.4863	.5433	.6006	.658	.716	.774	.832	.889	.947	1.090	1.233	1.376	1.520	1.810
20	**.0828**	**.1112**	**.1395**	**.1955**	**.2516**	**.3071**	**.3645**	**.4205**	**.4770**	**.5330**	**.5890**	**.645**	**.701**	**.757**	**.813**	**.869**	**.927**	**1.067**	**1.208**	**1.348**	**1.489**	**1.770**
30	.0811	.1088	.1366	.1916	.2465	.3015	.3570	.4121	.4672	.5221	.5771	.632	.687	.742	.797	.852	.908	1.046	1.184	1.322	1.460	1.735
40	.0795	.1067	.1338	.1876	.2415	.2954	.3503	.4038	.4576	.5114	.5652	.619	.673	.727	.781	.835	.890	1.025	1.161	1.296	1.431	1.701
50	.0780	.1045	.1310	.1839	.2367	.2905	.3432	.3960	.4487	.5014	.5541	.607	.660	.713	.766	.819	.873	1.006	1.139	1.271	1.403	1.668
60	.0764	.1025	.1283	.1803	.2323	.2840	.3562	.3882	.4402	.4927	.5447	.596	.649	.700	.752	.804	.856	.988	1.116	1.245	1.376	1.636
70	**.0760**	**.1005**	**.1260**	**.1770**	**.2280**	**.2791**	**.3302**	**.3808**	**.4316**	**.4824**	**.5332**	**.584**	**.635**	**.686**	**.737**	**.788**	**.839**	**.967**	**1.096**	**1.223**	**1.350**	**1.604**
80	.0736	.0988	.1239	.1738	.2237	.2739	.3242	.3738	.4234	.4729	.5224	.572	.622	.673	.723	.774	.824	.949	1.074	1.199	1.325	1.573
90	.0723	.0970	.1218	.1707	.2195	.2688	.3182	.3670	.4154	.4639	.5122	.561	.611	.660	.709	.759	.809	.932	1.054	1.177	1.300	1.544
100	.0710	.0954	.1197	.1676	.2155	.2638	.3122	.3602	.4079	.4555	.5033	.551	.599	.648	.696	.745	.794	.914	1.035	1.155	1.276	1.517
110	.0698	.0937	.1176	.1645	.2115	.2593	.3070	.3542	.4011	.4481	.4950	.542	.589	.637	.685	.732	.780	.899	1.017	1.135	1.254	1.491
120	**.0686**	**.0921**	**.1156**	**.1618**	**.2080**	**.2549**	**.3018**	**.3481**	**.3944**	**.4403**	**.4866**	**.533**	**.579**	**.626**	**.673**	**.720**	**.767**	**.884**	**1.001**	**1.118**	**1.234**	**1.465**
130	.0674	.0905	.1135	.1590	.2045	.2505	.2966	.3446	.3924	.4296	.4770	.524	.570	.616	.662	.708	.754	.869	.984	1.099	1.214	1.440
140	.0663	.0889	.1115	.1565	.2015	.2465	.2915	.3364	.3813	.4262	.4711	.516	.561	.606	.651	.696	.742	.855	.968	1.081	1.194	1.416
150	.0652	.0874	.1096	.1541	.1985	.2425	.2865	.3308	.3751	.4193	.4636	.508	.552	.596	.640	.685	.730	.841	.953	1.064	1.175	1.392
175	.0626	.0840	.1054	.1482	.1910	.2335	.2755	.3181	.3607	.4033	.4450	.488	.531	.573	.616	.658	.701	.808	.914	1.021	1.128	1.337
200	**.0603**	**.0809**	**.1014**	**.1427**	**.1840**	**.2248**	**.2655**	**.3064**	**.3473**	**.3882**	**.4291**	**.470**	**.611**	**.652**	**.592**	**.633**	**.674**	**.776**	**.879**	**.982**	**1.084**	**1.287**
225	.0581	.0779	.0976	.1373	.1770	.2163	.2555	.2949	.3344	.3738	.4129	.452	.491	.531	.570	.609	.649	.747	.846	.944	1.043	1.240
250	.0560	.0751	.0941	.1323	.1705	.2085	.2466	.2845	.3223	.3602	.3981	.436	.474	.513	.551	.589	.627	.722	.817	.912	1.007	1.197
275	.0541	.0726	.0910	.1278	.1645	.2011	.2378	.2745	.3111	.3478	.3844	.421	.458	.494	.531	.568	.605	.697	.789	.881	.972	1.155
300	.0523	.0707	.0881	.1237	.1592	.1945	.2300	.2654	.3008	.3362	.3716	.407	.442	.478	.513	.549	.585	.673	.762	.852	.940	1.118
350	**.0491**	**.0658**	**.0825**	**.1160**	**.1496**	**.1828**	**.2160**	**.2492**	**.2824**	**.3156**	**.3488**	**.382**	**.416**	**.449**	**.482**	**.616**	**.649**	**.632**	**.716**	**.799**	**.883**	**1.048**
400	.0463	.0621	.0779	.1090	.1405	.1720	.2035	.2348	.2661	.2974	.3287	.360	.391	.423	.454	.486	.517	.596	.674	.753	.831	.987
450	.0437	.0586	.0735	.1033	.1330	.1628	.1925	.2220	.2515	.2810	.3105	.340	.369	.399	.429	.458	.488	.562	.637	.711	.786	.934
500	.0414	.0555	.0696	.0978	.1260	.1540	.1820	.2100	.2380	.2660	.2940	.322	.351	.379	.407	.435	.463	.534	.604	.675	.746	.885
550	.0394	.0528	.0661	.0930	.1198	.1464	.1730	.1996	.2262	.2528	.2794	.306	.333	.359	.386	.413	.440	.507	.573	.641	.709	.841
600	.0376	.0504	.0631	.0885	.1140	.1395	.1650	.1904	.2158	.2412	.2668	.292	.317	.343	.368	.393	.419	.483	.547	.611	.675	.801

TABLES

TABLE II.—VOLUME IN CUBIC FEET OF 1 LB. OF AIR AT ATMOSPHERIC PRESSURE AT SEA LEVEL AND AT VARIOUS TEMPERATURES

Degrees Fahr.	Volume at atmos. pressure		Degrees Fahr.	Volume at atmos. pressure	
	Cubic feet in 1 lb.	Comparative volume		Cubic feet in 1 lb.	Comparative volume
0	11.583	.881	130	14.846	1.130
32	12.387	.943	140	15.100	1.149
40	12.586	.958	150	15.351	1.168
50	12.840	.977	160	15.603	1.187
62	13.141	1.000	170	15.854	1.206
70	13.342	1.015	180	16.106	1.226
80	13.593	1.034	200	16.606	1.264
90	13.845	1.054	210	16.860	1.283
100	14.096	1.073	212	16.910	1.287
110	14.344	1.092	220	17.128	1.301
120	14.592	1.111			

TABLE III.—VOLUMES, MEAN PRESSURES PER STROKE, AND FINAL TEMPERATURES IN AIR COMPRESSION AT SEA LEVEL

(Initial temperature = 60° Fahr.)

1	2	3	4	5	6	7	8	9
Gage pressure	Absolute pressure	Atmospheres	Volume of air isothermal compression	Volume of air adiabatic compression	Mean pressure per stroke isothermal compression	Mean pressure per stroke adiabatic compression	Final temperature, degrees Fahr. adiabatic compression	Gage pressure
0	14.7	1.	1.	1.	0.	0.	60	0
5	19.7	1.34	.7462	.81	4.3	4.5	106	5
10	24.7	1.68	.5952	.69	7.62	8.27	145	10
15	29.7	2.02	.495	.606	10.33	11.51	178	15
20	34.7	2.36	.4237	.543	12.62	14.4	207	20
25	39.7	2.7	.3703	.494	14.59	17.01	234	25
30	44.7	3.04	.3289	.4638	16.34	19.4	255	30
35	49.7	3.381	.2957	.42	17.92	21.6	281	35
40	54.7	3.721	.2687	.393	19.32	23.66	302	40
45	59.7	4.061	.2462	.37	20.52	25.59	321	45
50	64.7	4.401	.2272	.35	21.79	27.39	339	50
55	69.7	4.741	.2109	.331	22.77	29.11	357	55
60	74.7	5.081	.1968	.3144	23.84	30.75	375	60
65	79.7	5.423	.1844	.301	24.77	31.69	389	65
70	84.7	5.762	.1735	.288	26.	33.73	405	70
75	89.7	6.102	.1639	.276	26.65	35.23	420	75
80	94.7	6.442	.1552	.267	27.33	36.6	432	80
85	99.7	6.782	.1474	.2566	28.05	37.94	447	85
90	104.7	7.122	.1404	.248	28.78	39.18	459	90
95	109.7	7.462	.134	.24	29.53	40.4	472	95
100	114.7	7.802	.1281	.232	30.07	41.6	485	100
105	119.7	8.142	.1228	.2254	30.81	42.78	496	105
110	124.7	8.483	.1178	.2189	31.39	43.91	507	110
115	129.7	8.823	.1133	.2129	31.98	44.98	518	115
120	134.7	9.163	.1091	.2073	32.54	46.04	529	120
125	139.7	9.503	.1052	.202	33.07	47.06	540	125
130	144.7	9.843	.1015	.1969	33.57	48.1	550	130
135	149.7	10.183	.0981	.1922	34.05	49.1	560	135
140	154.7	10.523	.095	.1878	34.57	50.02	570	140
145	159.7	10.846	.0921	.1837	35.09	51.	580	145
150	164.7	11.204	.0892	.1796	35.48	51.89	589	150

TABLE V.—THEORETICAL HORSE-POWER AND FINAL TEMPERATURES
(Initial temperature = 60° Fahr. at sea level)

Gage pressures	Atmospheres	Single-stage compression				Two-stage compression			Three-stage compression			Four-stage compression		
		Iso-thermal	Adiabatic											
		Horse-power required to compress and deliver 1 cu. ft. free air per minute	Horse-power required to compress and deliver 1 cu. ft. free air per minute	Efficiency as compared to isothermal	Final temperature degrees Fahr.	Horse-power required to compress and deliver 1 cu. ft. free air per minute	Efficiency as compared to isothermal	Final temperature degrees Fahr. perfect inter-cooling adiabatic compression	Horse-power required to compress and deliver 1 cu. ft. free air per minute	Efficiency as compared to isothermal	Final temperature degrees Fahr. perfect inter-cooling adiabatic compression	Horse-power required to compress and deliver 1 cu. ft. free air per minute	Efficiency as compared to isothermal	Final temperature degrees Fahr. perfect inter-cooling a diabatic compression
5	1.34	.0188	.0197	.96	106	....	....	....	....	....	....	....	....	....
10	1.68	.0333	.0362	.93	145	....	....	....	....	....	....	....	....	....
15	2.02	.0481	.0505	.90	178	....	....	....	....	....	....	....	....	....
20	2.36	.0551	.0630	.88	207	....	....	....	....	....	....	....	....	....
25	2.70	.0638	.075	.85	234	....	....	....	....	....	....	....	....	....
30	3.04	.0713	.085	.84	252	....	....	....	....	....	....	....	....	....
40	3.72	.0843	.104	.81	302	....	....	....	....	....	....	....	....	....
50	4.40	.0948	.120	.79	339	.109	.87	188	....	....	....	....	....	....
60	5.08	.1037	.134	.77	375	.121	.86	203	....	....	....	....	....	....
70	5.76	.1120	.148	.75	405	.131	.85	214	....	....	....	....	....	....
80	6.44	.1196	.16	.74	432	.141	.85	224	....	....	....	....	....	....
90	7.12	.1260	.171	.74	459	.150	.84	234	....	....	....	....	....	....
100	7.80	.1320	.182	.73	485	.158	.83	243	....	....	....	....	....	....
110	8.48	.1371	.192	.72	500	.165	.83	250	....	....	....	....	....	....
120	9.16	.1422	.202	.71	529	.172	.83	257	....	....	....	....	....	....
130	9.84	.1467	.210	.70	560	.179	.82	265	....	....	....	....	....	....
140	10.52	.1510	.218	.69	570	.186	.82	272	....	....	....	....	....	....
150	11.20	.1547	.226	.69	589	.193	.81	279	.182	.85	200	....	....	....
160	11.88	.1583	.234	.68	607	.198	.81	285	.187	.85	204	....	....	....
180	13.24	.1656	.249	.67	640	.208	.80	297	.197	.84	211	....	....	....
200	14.60	.1720	.263	.65	672	.217	.79	309	.206	.83	218	....	....	....
225	16.3	.1790	.278	.64	715	.227	.79	320	.215	.83	224	....	....	....
250	18.	.1860	.292	.64	749	.237	.78	331	.224	.83	230	....	....	....
275	19.7	.1920	.306	.63	780	.247	.78	342	.233	.82	236	....	....	....
300	21.4	.1970	.317	.62	815	.256	.77	352	.241	.82	241	....	....	....
350	24.8	.2060	.342	.60	867	.272	.76	370	.252	.82	250	....	....	....
400	28.2	.2140	.364	.59	915	.283	.76	380	.262	.82	258	....	....	....
450	31.6	.2230	.381	.58	960	.295	.75	397	.272	.82	266	....	....	....
500	35.	.2290	.398	.57	1000	.307	.75	415	.282	.81	275	.26	.88	215
550	38.4	.2340	.416	.56	1040	.321	.73	430	.292	.80	283	.269	.87	220
600	41.8	.240	.432	.55	1077	.332	.72	442	.300	.80	290	.278	.86	225
650	45.2	.245	.447	.55	1113	.345	.71	451	.31	.79	295	.284	.86	228
700	48.6	.249	.461	.54	1136	.355	.70	458	.32	.78	300	.29	.86	234
750	52.	.252	.475	.53	1178	.363	.69	462	.327	.78	305	.296	.85	236
800	55.4	.258	.488	.52	1208	.373	.69	468	.334	.78	309	.302	.85	240
850	58.8	.262	.500	.52	1237	.381	.69	480	.341	.77	314	.307	.85	244
900	62.2	.265	.512	.52	1265	.388	.68	490	.347	.76	319	.312	.85	247
950	65.6	.268	.523	.51	1292	.395	.68	495	.354	.76	322	.316	.85	250
1000	69.	.272	.534	.51	1318	.403	.67	498	.360	.75	325	.32	.85	252
1100	75.8	.278	.555	.50	1367	.416	.67	507	.370	.75	331	.327	.85	254
1200	82.6	.283	.575	.49	1415	.429	.66	525	.381	.74	338	.334	.84	258
1300	89.4	.289	.594	.49	1457	.441	.66	534	.390	.74	342	.341	.84	265
1400	96.2	.293	.611	.48	1498	.452	.65	550	.399	.74	349	.348	.84	270
1500	103.	.297	.627	.48	1537	.462	.65	563	.406	.73	353	.355	.84	273
1600	109.8	.301	.643	.47	1575	.472	.64	568	.415	.73	358	.361	.83	276
1700	116.6	.305	.659	.47	1610	.482	.63	589	.424	.72	364	.367	.83	280
1800	123.4	.309	.673	.46	1645	.491	.63	606	.431	.72	370	.372	.83	284
1900	130.2	.313	.687	.46	1678	.500	.63	628	.438	.72	374	.377	.83	287
2000	139.	.317	.701	.45	1709	.509	.62	639	.444	.71	378	.381	.83	290
2250	154.	.324	.733	.44	1784	.528	.62	645	.460	.70	385	.393	.82	294
2500	171.	.331	.763	.43	1852	.547	.61	654	.474	.70	398	.405	.82	298
3000	205.	.342	.816	.42	1975	.579	.59	670	.500	.69	414	.42	.81	308
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15

TABLE IV.—VOLUME WHICH 1 CU. FT. OF FREE AIR, HAVING A TEMPERATURE OF 60° FAHR., WILL OCCUPY WHEN COMPRESSED IN ONE STAGE ADIABATICALLY TO VARIOUS ATMOSPHERES

Also final temperature of the air at such pressures

1	2	3	4
Pressure in atmospheres	Absolute pressures in lb. per. sq. in.	Volumes in cu. ft. adiab. comp.	Final temp., degrees Fahrenh.
1.00	14.70	1.000	60.0
1.25	18.37	0.854	94.8
1.50	22.05	0.750	124.9
2.00	29.40	0.612	175.8
2.50	36.70	0.522	218.3
3.00	44.10	0.459	255.1
3.50	51.40	0.411	287.8
4.00	58.80	0.374	317.4
5.00	73.50	0.319	369.4
6.00	88.20	0.281	414.5
7.00	102.90	0.252	454.5
8.00	117.60	0.229	490.6
9.00	132.30	0.211	523.7
10.00	147.00	0.195	554.0
15.00	220.50	0.147	681.0

TABLE VI.—MULTIPLIERS FOR DETERMINING THE VOLUME OF FREE AIR AT VARIOUS ALTITUDES WHICH, WHEN COMPRESSED TO VARIOUS PRESSURES, IS EQUIVALENT IN EFFECT TO A GIVEN VOLUME OF FREE AIR AT SEA LEVEL

Altitude in feet	Barometric pressure		Multiplier				
	Inches of mercury	Pounds per square inch	Gage pressure (pounds)				
			60	80	100	125	150
0	30.00	14.75	1.000	1.000	1.000	1.000	1.000
1,000	28.88	14.20	1.032	1.033	1.034	1.035	1.036
2,000	27.80	13.67	1.064	1.066	1.068	1.071	1.072
3,000	26.76	13.16	1.097	1.102	1.105	1.107	1.109
4,000	25.76	12.67	1.132	1.139	1.142	1.147	1.149
5,000	24.79	12.20	1.168	1.178	1.182	1.187	1.190
6,000	23.86	11.73	1.206	1.218	1.224	1.231	1.234
7,000	22.97	11.30	1.245	1.258	1.267	1.274	1.278
8,000	22.11	10.87	1.287	1.300	1.310	1.319	1.326
9,000	21.29	10.46	1.329	1.346	1.356	1.366	1.374
10,000	20.49	10.07	1.373	1.394	1.404	1.416	1.424

TABLE VII.—EFFECT OF INITIAL OR IN-TAKE TEMPERATURE ON EFFICIENCY AND CAPACITY OF AIR COMPRESSORS

Unit capacity and efficiency assumed at 60° Fahr.

Initial temperature		Relative capacities and efficiencies	Initial temperature		Relative capacities and efficiencies
Degrees Fahr.	Degrees Abs.		Degrees Fahr.	Degrees Abs.	
−20	441	1.18	70	531	0.980
−10	451	1.155	80	541	0.961
0	461	1.13	90	551	0.944
10	471	1.104	100	561	0.928
20	481	1.083	110	571	0.912
30	491	1.061	120	581	0.896
32	493	1.058	130	591	0.880
40	501	1.040	140	601	0.866
50	511	1.020	150	611	0.852
60	521	1.000	160	621	0.838

Adiabatic Compression and Delivery:

TABLE VIII.—THEORETICAL H.P. FINAL VOLUME AND FINAL ABSOLUTE TEMPERATURE

No. of stages	Theoretical horse power	Final volume in cubic feet	Final absolute temperature
1 stage	$\frac{144nP_aV_a}{33{,}000(n-1)}\left[\left(\frac{P_1}{P_a}\right)^{\frac{n-1}{n}}-1\right]$	$V_1=V_a\left(\frac{P_a}{P_1}\right)^{\frac{1}{n}}=0.71$	$T_1=T_a\left(\frac{P_1}{P_a}\right)^{\frac{n-1}{n}}=0.29$
2 stage	$2\frac{144nP_aV_a}{33{,}000(n-1)}\left[\left(\frac{P_2}{P_a}\right)^{\frac{n-1}{2n}}-1\right]$	$V_2=V_a\left(\frac{P_a}{P_2}\right)^{\frac{n+1}{2n}}=0.856$	$T_2=T_a\left(\frac{P_2}{P_a}\right)^{\frac{n-1}{2n}}=0.144$
3 stage	$3\frac{144nP_aV_a}{33{,}000(n-1)}\left[\left(\frac{P_3}{P_a}\right)^{\frac{n-1}{3n}}-1\right]$	$V_3=V_a\left(\frac{P_a}{P_3}\right)^{\frac{2n+1}{3n}}=0.9037$	$T_3=T_a\left(\frac{P_3}{P_a}\right)^{\frac{n-1}{3n}}=0.0963$
4 stage	$4\frac{144nP_aV_a}{33{,}000(n-1)}\left[\left(\frac{P_4}{P_a}\right)^{\frac{n-1}{4n}}-1\right]$	$V_4=V_a\left(\frac{P_a}{P_4}\right)^{\frac{3n+1}{4n}}=0.9278$	$T_4=T_a\left(\frac{P_4}{P_a}\right)^{\frac{n-1}{4n}}=0.0722$

TABLE IX.—FOR DETERMINING DIAMETERS OF BRANCH PIPES

Relative carrying capacity of pipes for air

Diam. of pipe	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	10	12
1	1.00	0.52	0.327	0.15	0.084	0.05	......	......	......	......	......	......	......	......	......
1¼	1.90	1.00	0.614	0.28	0.16	0.10	0.066	......	......	......	......	......	......	......	......
1½	3.05	1.60	1.00	0.46	0.256	0.16	0.106	0.075	......	......	......	......	......	......	......
2	6.55	3.45	2.14	1.00	0.56	0.34	0.23	0.160	0.12	......	......	......	......	......	......
2½	11.8	6.25	3.88	1.81	1.00	0.614	0.41	0.29	0.216	0.163	......	......	......	......	......
3	19.0	12.0	6.32	2.95	1.63	1.00	0.67	0.47	0.35	0.268	0.165	......	......	......	......
3½	......	15.2	9.45	4.3	2.43	1.50	1.00	0.71	0.52	0.4	0.246	0.166	......	......	......
4	......	21.6	13.4	6.25	3.46	2.10	1.42	1.00	0.75	0.56	0.352	0.237	0.169	......	......
4½	......	......	18.0	8.30	4.65	2.85	1.90	1.35	1.00	0.76	0.475	0.32	0.227	0.128	......
5	......	......	......	......	6.14	3.77	2.60	1.78	1.32	1.00	0.625	0.42	0.30	0.169	0.10
6	......	......	......	......	......	6.05	4.00	2.85	2.15	1.60	1.00	0.675	0.48	0.27	0.17
7	......	......	......	......	......	......	6.00	4.20	3.16	2.37	1.48	1.00	0.71	0.40	0.25
8	......	......	......	......	......	......	......	6.00	4.40	3.25	2.10	1.40	1.00	0.56	0.35
10	......	......	......	......	......	......	......	......	7.85	5.90	3.70	2.50	1.77	1.00	0.63
12	......	......	......	......	......	......	......	......	......	9.40	5.90	3.95	2.80	1.60	1.00

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